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AND
TRACTORS

A Textbook for Students and
A Handbook for Engineers

BY

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Member, Society of Automotive Engineers

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PREFACE

THE present volume was written to take the place of Vol. II of *The Gasoline Automobile,* of which latter four editions were printed between 1912 and 1920. The chief reason for issuing it under a new title, rather than calling it a new edition of the old volume, is that the scope of the work has been changed somewhat. The old volume dealt with both design and production methods, while the new deals with design only. But while the scope thus has been narrowed in one respect, it has been broadened in another, in that more attention is given to heavy commercial vehicles and that agricultural tractors also are dealt with. Another reason justifying the new title is that although a certain amount of material, notably mathematical discussion of fundamental relations, has been carried over from the old volume, the great bulk of the material is new.

Several reasons prompted the author to omit production methods from the work. Such methods have seen great development in recent years, and it would have been almost impossible to do them justice in the space which could have been devoted to them in a book dealing primarily with design. Moreover, the practices in vogue are changing rapidly, which makes the subject more suitable for treatment in periodical publications, and these are covering it quite thoroughly.

The book is intended as a textbook for students taking courses in automotive engineering, for home-study by those engaged in automotive work of a technical character or wishing to prepare themselves for such work, and for reference purposes in designing new automotive products. The old volume was widely used for all of these purposes, and it is the author's fond hope that the new one will meet equally well the requirements of these different classes of reader.

In preparing the text it has been the aim, wherever possible, to trace the relations of mechanisms analysed back to fundamental principles, with a view to giving the student a clear insight into the problem in hand, and to enable him to apply similar methods to other problems that may arise in the course of his engineering work. The man who has a thorough knowledge of the principles of the sciences under-

lying his particular branch of engineering is naturally much better qualified to do effective work of a developmental and research character than he whose only recourse is search of reference literature for information bearing directly on his problem.

The same general division of the material as in the old volume has been retained; however, several new chapters have been introduced, on subjects that have attained importance recently, while others, dealing with subjects that are now more or less obsolete (friction transmissions, planetary transmissions), have been eliminated.

One chapter is given over to a discussion of the principles of toothed gearing. This may seem a rather general subject for a work dealing with a highly specialized line of engineering work, but it was thought that it would help those not already familiar with it in following the discussions in chapters devoted to applications of such gearing in automotive vehicles, such as those on Change-Speed Gears, The Differential Gear, Bevel-Gear-Driven Rear Axles, etc.

The six-wheeled commercial vehicle chassis has assumed considerable importance since 1920, and is dealt with in a chapter which also covers front-wheel drives and four-wheel drives. Another chapter deals with gas-electric drives, which have achieved a considerable degree of popularity with operators of large fleets of buses, and are also much used on large rail cars.

Independent suspension is a subject that has been much to the fore in Europe in recent years, and though it is not yet used in production vehicles in this country, it was considered opportune to devote a chapter to it. Such suspension not only improves the riding qualities of vehicles, but suppresses certain forms of vibration, and its introduction in American practice is probably only a matter of time.

The chapter on Agricultural Tractors deserves particular mention, in that it considerably widens the scope of the work. This branch of the automotive industry passed through a period of stagnancy following the industrial depression of 1920, but recently it has shown signs of renewed activity, so that it will undoubtedly offer opportunities for increased engineering work in the future. Since most of the parts of farm tractors are similar, functionally, to corresponding parts of motor trucks, the treatment is made brief, but the usual forms of construction are outlined, and experience constants are given for most of the more important working parts.

American motorcars have long been manufactured on the interchangeable plan, and as long ago as 1908 the Cadillac Motor Car Company was awarded a prize by the Royal Automobile Club of Great Britain for the degree to which the parts of its cars were interchangeable. However, the principles governing the setting of allowances and tolerances for parts in interchangeable manufacture have been definitely formulated only during the past decade, and since the designer must assign values to these items and enter them on his drawings, it was thought well to add a chapter covering the latest developments along these lines.

The author wishes here to express his indebtedness to the editors of *Automotive Industries* and of *The Automobile Engineer* for permission to make use of a number of illustrations which appeared first in the columns of these periodicals. Mention should be made also of the fact that contributions to the Transactions of the Society of Automotive Engineers and the Institution of Automobile Engineers have been drawn upon freely.

CONTENTS

CHAPTER		PAGE
I.	Types of Vehicle and Their General Arrangement	1
II.	Friction Clutches	14
III.	Some Principles of Toothed Gearing	42
IV.	Change-speed Gears	67
V.	Propeller Shafts and Universal Joints	121
VI.	The Differential Gear	151
VII.	Bevel-gear-driven Rear Axles	168
VIII.	The Worm Drive	219
IX.	Double-reduction Axles	245
X.	Front-drive, Four-wheel-drive and Six-wheeled Vehicles	263
XI.	Brakes	280
XII.	Front Axles	348
XIII.	The Steering Gear	375
XIV.	Control	410
XV.	Frames and Their Brackets	430
XVI.	Chassis Springs	453
XVII.	Independent Suspensions	498
XVIII.	Road Wheels	509
XIX.	Chassis Lubrication	534
XX.	Gas-electric Drives	547
XXI.	The Chain Drive	564
XXII.	Some Problems of Tractor Design	575
XXIII.	Fits and Tolerances	615
XXIV.	Methods of Rust Protection	629
	Appendix	641
	Index	673

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CHAPTER I

Types of Vehicle and Their General Arrangement

Development of the internal combustion engine to a point of high weight efficiency led to its application to different types of cycle, road vehicle, tractor, locomotive and railcar. Types now in commercial production include the following:

1. The motorcycle, a two-wheeled machine which is used in large numbers in Europe, but is relatively less popular in this country. It has the advantages of low fuel and upkeep cost, and of compactness, which latter enables it to be stored almost anywhere, but it is necessarily less comfortable than the automobile.

2. The (private) passenger automobile, the most popular type of motor vehicle, of which more than 20,000,000 are in use in the United States alone; it is built for from two to seven passengers.

3. The taxicab, a type for commercial transportation, generally quite similar to the private passenger car, but when specially designed for the purpose, usually more staunch and less speedy.

4. The motor bus or motor coach, for the commercial transportation of passengers in larger numbers.

5. The motor truck, for the transportation of merchandise over the roads. In the past it has been customary to regard the delivery wagon, a commercial vehicle of less than 1-ton load capacity, as a separate type, but the line of demarkation is being wiped out.

6. The road tractor or tractor-truck, a vehicle very similar in general design to a truck, but of shorter wheelbase and intended to haul a semi-trailer.

7. The industrial tractor, a low-wheeled machine, intended for use in factory yards, on steamship piers and on railroad platforms. It has been equipped mostly with electric motors, but gasoline-powered types are also in production.

8. The farm tractor, a machine designed chiefly for plowing and other farm work, but also used for industrial pur-

poses. It operates at low speeds and delivers most of the engine power through a drawbar.

9. The industrial locomotive, for use on narrow-gauge tracks, to take the place of steam locomotives formerly used for the purpose.

10. The railcar, a motorized passenger coach for operation on rails, used by the railroads particularly on branch lines on which there is comparatively little traffic.

11. Various types of special self-propelled machines, such as pieces of fire department equipment, road building and road maintenance machines, etc.

The general principles discussed in this volume apply to all of the types enumerated above. All of them have power transmission and control mechanism, and brakes; all road vehicles have steering gears, and the majority of the types have frames, and springs supporting these frames. In dealing with the application of general principles to practical design, motor road vehicles, that is, passenger cars, trucks and buses, have been kept in view chiefly, but a few applications to other types also are given.

Chassis and Body—Practically all motor vehicles comprise two distinct parts, the chassis or running gear, and the body. The chassis by itself forms a sort of road locomotive, comprising all of the elements necessary to propel itself over the roads, while the body is designed to receive the load to be carried. In the case of passenger cars the body is designed and equipped to seat the passengers in comfort and to afford them protection against wind and weather. Although the chassis and body can be built in one unit, and building them so results in an appreciable saving in weight, in practice it has been found preferable to make them separate parts. This facilitates manufacturing operations, because chassis of the same design can be equipped with bodies of different types, and all of the chassis therefore can be assembled on the same line, which makes for economy. Moreover, the manufacturing processes employed in the production of the two units are rather different, and chassis and body production therefore can be carried on to advantage in different plants, or in different sections of the same plant. From the standpoint of ease of maintenance also there is an advantage in the separation of chassis and body, for in the case of important repairs or overhauls the body can be removed from the chassis, which makes the chassis units more accessible, and prevents injury to the body and the soiling of its upholstery by the mechanics while working on greasy chassis parts.

Number of Road Wheels—The chassis is built with a

frame on which the engine and the power transmission and control units are assembled. In the conventional motor vehicle this frame is supported on two axles through the intermediary of chassis springs, and each axle is supported by two road wheels. In the early years of the industry numerous three-wheeled vehicles were developed, most of them having a single steering wheel in front and two driving wheels at the rear. But the buying public did not take kindly to the three-wheeler, which has the disadvantages that it is less stable than a four-wheeler, and that on rutted roads the steering wheel has to run on the rough part of the road between tracks. Reasons of an esthetic nature also probably had a good deal to do with the public's apathy toward the three-wheeled construction.

Steering and Driving Wheels—With four road wheels, two front and two rear, the question arises as to which shall be used for steering and which for driving. It is, of course, possible to use all four for both steering and driving, but this involves considerable complication and is unnecessarily expensive in manufacture. Steering by the rear wheels, moreover, involves a serious disadvantage, for if a vehicle with rear-wheel steering is parked alongside a curb it can gain the middle of the road only by backing, for if it is started forward the rear wheels will run into the curb. This defect is inherent in both rear-wheel and four-wheel steering, for which reason all or nearly all motor-propelled road vehicles are steered by their front wheels.

Sufficient traction can be obtained with two driving wheels under nearly all conditions of traffic on "made" roads. Of course, when the roads are icy or covered deeply with snow, it is necessary to use chains or other traction devices. As it is simpler to transmit the driving power through the non-steering than through the steering wheels, driving is effected practically always through the rear wheels. To make the rear wheels most effective as drivers, the design is so laid out that more than half of the total weight is carried by them, for the limiting traction obtainable from a wheel depends upon the adherence of the wheel or its tire to the road surface, which in turn is directly proportional to the weight carried by the wheel. From this point of view it is fortunate that application of the propelling force tends to raise the front wheels and to press the rear wheels more forcibly against the road surface, thus increasing the adherence of the latter. The greater the propelling effort, the greater the transfer of weight from the front to the rear wheels and the greater the adherence of the rear (driving) wheels to the road.

There is a similar effect due to grades. The greatest traction usually is needed in ascending steep grades, and on such grades, owing to the inclination of the vehicle, the center of gravity is shifted toward the rear, and the load on the rear wheels is increased.

While this transfer of weight is advantageous in that it increases the adherence of the driving wheels, it must not be carried too far, for it simultaneously reduces the ground pressure of the front wheels, on which effective steering depends. The ability of the front wheels to direct the course of the vehicle is dependent on the fact that their resistance to rolling motion is less than their resistance to sideward sliding motion; but the difference between these two resistances is directly proportional to the force with which the wheels press against the ground, and if this force is reduced to zero the steering power of the front wheels also is reduced to zero.

Four-Wheel Drive—In cross-country operation it is difficult to get enough traction on two wheels and still retain enough load on the steering wheels to assure positive steering under all conditions. For such work four-wheel drive vehicles have advantages. During the World War the Ordnance Department of the U. S. Army, whose vehicles had to operate over the shell-torn areas close to the firing lines, used four-wheel drive trucks almost exclusively, and these vehicles are used also for commercial purposes, particularly in rough country. With four-wheel drive the entire weight of the vehicle and load is available for traction purposes, and the weight can be divided substantially equally between the front and rear wheels, which results in positive steering under practically all conditions and in more nearly equalized tire wear.

Securing the greatest possible ground adherence and traction is of particular importance in connection with tractors, which not only must propel their own weight but must develop a large drawbar pull besides. Farm tractors therefore are so designed that when the engine is developing its maximum torque and is driving through the largest reduction gear practically all of the weight rests on the driving wheels. This is the condition under which the engine develops approximately the maximum torque on the driving wheels, and the corresponding ground adherence therefore should be a maximum. Since under this condition nearly all of the weight of the tractor is available for traction, little could be gained by using a four-wheel drive, and four-wheel drive farm tractors never have found extensive application. The additional

cost of production is not warranted by the gain in the limiting traction. However, in the so-called crawler type of farm tractor, which runs on movable chain tracks that are moved along by the tractor itself by means of chain wheels, the entire weight is available for traction purposes.

Frames—Frames for motor vehicles are generally made of pressed steel members. There are two side rails and a variable number of cross members, the different parts being united by riveting, as a rule. In the earliest pressed steel frames the top edge of the side rails usually lay in a horizontal plane, but in modern passenger-car chassis, in order to bring the body closer to the ground and thus lower the center of gravity of the car as a whole, thereby increasing its stability, the side rails are usually made with a drop a short distance ahead of the rear axle and often with another drop some distance behind the front axle. The frames of buses also are lowered by means of drops or kick-ups in the side rails, both to make the vehicles safer at high speeds and to facilitate ingress and egress of passengers. Truck frames, however, must be of such height that the height of the body floor corresponds to the level of loading platforms, and the top edges of truck-frame side rails are made straight.

Passenger car frames must be made wide at the rear, to correspond to the width of the body, while at the front they must be made narrow to permit of a large steering lock, which means a short minimum turning radius. To meet these requirements it was formerly the practice to offset the frame side rails at about the point where the dash is located, but it is now customary to distribute the increase in frame width over the whole or nearly the whole length of the frame.

Engine Location—The pioneers of the automobile considered it necessary, in order not to offend the esthetic sense of prospective buyers, to locate the engine and driving mechanism under the seat or seats of the body, and in general to adhere as closely as possible to the outward form of horse vehicles. Experience, however, showed this plan to be impracticable, because the passengers had to be disturbed whenever it was necessary to inspect some part of the engine or to make the slightest adjustment. The powerplant, moreover, was generally quite inaccessible, and with increasing powers the heat from the engine became objectionable in summer time. It is now the almost universal practice to place the engine on the chassis at the forward end of the latter, under a sheet-metal, hinged hood, which can be raised to give access to the mechanism. At the rear the engine space is closed in by a partition referred to as the dash (a term

carried over from carriage practice), while the forward wall of the space is constituted by the radiator. The body begins with the dash.

In commercial vehicles, where it is desirable to utilize the available space to the utmost, the driver's seat sometimes is arranged over or alongside of the engine, and in one type of bus, known as the street-car type, the old plan of placing the powerplant under the seats, has been reverted to, for the sake of a gain in seating capacity. The greatly improved reliability of the modern engine practically entirely eliminates the objections formerly urged against this arrangement, and the use of the so-called gas-electric drive, quite common in this type of bus, makes it practical to place the powerplant at any convenient point of the chassis, as the problem of connecting it up mechanically with the driving axle does not arise.

The most forward section of the floor of the body, known as the toeboard, slopes upward toward the dash, and under it is located the flywheel and clutch housing. From this point back no part of the chassis except control members should project above the top plane of the frame.

Friction Clutch—Owing to the fact that the gasoline engine—unlike the steam engine and electric motor—does not start from a standstill with full torque, but must be started by means of a hand crank or an auxiliary starting motor, which latter generally produces sufficient torque only to turn the engine over against compression, it is necessary to provide a device whereby the engine can be disconnected from the driving parts of the vehicle for starting it, and can be connected to them again after it has attained speed. The device used for this purpose, known as the friction clutch, permits its parts to slip relative to each other, until the engine speed has been reduced and the vehicle brought up to a speed corresponding to the engine speed. The clutch is always placed close to the engine and is usually incorporated in the flywheel.

Change-Speed Gear—Owing to the limited flexibility of the gasoline engine, it is necessary to provide means for connecting the engine to the driving wheels in several different ratios. The gasoline engine differs from other light motors in that when running at its speed of maximum economy or at its speed of maximum output, it is producing nearly the maximum torque of which it is capable. The engine, of course, must be so proportioned and geared that under normal conditions of operation—that is, when the car is traveling over level roads at good speed—it runs at about its speed of

maximum economy, and it is then incapable of delivering the propelling effort required for climbing steep hills or passing through deep sand, through the same gear reduction.

The reader undoubtedly is familiar with the fact that when two shafts are connected by gearing, the torques (or turning moments) of the two bear to each other the inverse relation of their respective speeds. Thus, by providing a hill-climbing gear giving a speed reduction, say, four times as great as the normal speed reduction, the driving effort at the caused by punctures and blow-outs of pneumatic tires. climbing or low gear gives a very low speed, it is customary in all but the lightest vehicles to have one or more intermediate speeds, for use on moderate hills or on soft or uneven roads. For passenger cars three-speed and four-speed gears are used, while for truck work, transmissions are made with as many as seven forward speeds, because in the case of pneumatic-tired trucks it is desired to be able to travel at passenger-car speeds when light, while at the same time the maximum driving effort must be sufficient to enable the truck to get out of an excavation, for instance, with a full load. This calls for a range of gear ratios of about 1 to 8, which it is impossible to cover adequately with only four changes, as gear shifting is difficult if individual steps are too large.

In addition to the forward speeds, at least one reverse motion is required. Transmissions with more than four forward speeds generally provide two reverse speeds, and in some applications of trucks, notably in road building, it has been found desirable to make one of the reverse speeds fairly high, as it is often necessary to back long distances, which can then be done with less loss of time.

A variable transmission is necessary also to make rapid acceleration from a standstill possible. The acceleration of the vehicle is directly proportional to the excess of engine power over that required to maintain the momentary speed. For a given vehicle speed the engine will be turning over much faster, and therefore will develop much more power, if it drives through the large-reduction or low-speed gear than if it drives through the high-speed gear. The maximum acceleration of modern passenger cars is about 2.5 miles per hour per second (about 3.5 feet per second per second) in high gear, while in low gear it may be as high as 8 miles per hour per second. High acceleration is particularly desirable under conditions of modern traffic control by signal lights.

Direct Drive—Variable gear transmissions generally are designed with a direct drive for high speed and for normal

operation generally. The power is then carried directly through the transmission without any of its gears being under load. Under this condition the transmission operates at the highest efficiency, with a minimum of noise, and with the least wear and tear on its members. A speed reduction is necessary, however, between the transmission and the driving axle, because engines are designed to operate at quite high speeds, to get the most power from a given displacement or a given weight, and the crankshaft speed is always much higher than the required axle speed. This reduction is obtained by the "final drive" gear, which is usually combined with the driving axle, and may be obtained in a single step or in two steps. In the first case we speak of a single-reduction final drive; in the second, of a double-reduction drive.

Final-Drive Ratios—In passenger cars and other high-speed vehicles a single reduction is always used, as the reduction need not be great. For example, suppose that the car is to have a speed of 70 m.p.h. for an engine speed of 3200 r.p.m. and that the wheel diameter is 30 in. The rear axle speed then will be

$$\frac{70 \times 5280 \times 12}{60 \times 30 \times 3.14} = 785 \text{ r.p.m}$$

Hence the rear axle reduction ratio must be 3200 to 785 or 4.08 to 1, which is easily obtained in a single step.

On the other hand, a solid-tired truck may be geared to run at 20 m.p.h. for an engine speed of 1400 r.p.m. If the truck wheels are 40 in. in diameter the axle speed will be

$$\frac{20 \times 5280 \times 12}{60 \times 40 \times 3.14} = 168.3 \text{ r.p.m.}$$

Hence the reduction ratio of the final drive must be 1400 to 168.3 or 8.32 to 1. This can be obtained in a single step by worm gearing, but requires a double reduction if any other form of gearing is to be used.

Final Drive Types—With few exceptions all of the passenger cars built in the United States have a final drive by bevel gears, spiral bevel gears being used because they are more quiet in operation than the older straight bevel gears. A few passenger cars have a final drive by worm and worm wheel. Worm drive for passenger cars was first introduced in England early in the history of automobile development, and its outstanding advantage at that time was its greater quietness as compared with the straight bevel gear, the only

type of bevel gear then available. With the introduction of spiral bevel gears in 1912 this advantage of the worm drive was practically nullified, and at the present time the chief reason for worm-gear drive on passenger cars in the United States is that, in the undermounted form, it locates the propeller shaft at a lower level and permits of lowering the frame and the body of the car.

Commercial cars of the lighter and faster types also use the spiral bevel gear final drive principally. The heavier and slower types use either a worm drive or else a double reduction drive, which may be by bevel and spur gears, bevel and internal gears, or bevel gears and chain.

Propeller Shaft and Universal Joints—The engine and transmission gear are carried by the frame, which is supported on the axles through the intermediary of the chassis springs. This involves that, owing to spring action, there is constant change in the relative angular positions of the transmission and the driving axle; still, the power of the engine must be transmitted from the former to the latter smoothly and uniformly, and without producing excessive bearing loads. There are two methods of transmission from a spring-mounted change gear to an axle or to wheels below the springs, viz., by chains and by shaft. The former is now rarely used, and shaft drive is nearly universal. The shaft, however, must be provided with one or two universal joints and with a slip joint. The former serve to transmit power between shafts whose axes intersect but make an angle with each other that varies in service. The slip joint takes care of slight changes in the distance between the rear end of the transmission and the center of the rear axle.

Differential Gear—If both driving wheels were positively connected to a single source of motive power, they could not rotate at unequal speeds, as they must when the car is turning a corner, if the tires are not to slip on the road. If the wheels had to drive the car forward only, the problem could be solved by driving them through ratchet clutches, but since they must drive the car backward as well, it is necessary to incorporate a so-called differential gear in the drive, through which the driving torque is always equally divided between the two driving wheels, for both directions of motion. The differential gear enables the driving wheels to turn at different speeds, as required by the course followed or by any slight difference in their diameters. With four-wheel drive it is necessary, or at least desirable, to provide three differential gears, one between the front and rear axles and one between the two wheels of each axle.

Tires—All motor-propelled road vehicles are fitted with rubber tires, which are made in three general types, viz., solid, pneumatic and cushion, the latter being intermediate between the two former with respect to resiliency, the most important characteristic of rubber tires. Aside from their resiliency, rubber tires possess valuable characteristics in their great adherence to normal road surfaces and their silence in operation. Many attempts were made during the early years of the industry, when the pneumatic tires produced still had serious defects, to use other tire materials, such as steel, wood and indurated fabric, but in every case it was found that the disadvantages of the substitute material outweighed its advantages, which latter consisted in lower manufacturing cost and freedom from the kind of road troubles that were caused by punctures and blow-outs of pneumatic tires.

The early defects of the pneumatic tire have been overcome to a large extent, and this type is constantly increasing its share of the field. Up to the present there has been a constant increase in road speeds, and high road speeds call for the superior cushioning effect of the pneumatic. Passenger cars always have been fitted practically exclusively with pneumatic tires. The earlier motor buses carried solid tires for the most part, but recently the trend in bus construction has been strongly toward pneumatic tires, and practically all buses for interurban service are now fitted with them. In the truck field the use of pneumatic tires started with the smaller units, but it is gradually creeping up into the larger sizes. In addition to making higher speeds practicable, pneumatic tires on heavy vehicles have the advantage that they reduce the wear and tear on roads. Several European countries have placed a ban on solid tires, and present indications are that ultimately the pneumatic tire will be used almost exclusively for all kinds of vehicle using the public roads.

Brakes—With the high speeds of modern road vehicles and congestion in city streets it is essential to safety that the vehicles be equipped with effective brakes. Horse vehicles were equipped with brakes acting directly on the tires of the wheels, but with high-speed motor vehicles this principle is inapplicable, as the tires would be quickly destroyed. A type of brake similar to that previously employed on cranes is being used, consisting of a metal drum or disc connected to one of the rotating parts of the vehicle, to which can be applied a friction shoe or band supported by a structural part. The friction member is made to grip the rotating disc or drum and bring it to a stop gradually, and in the same

measure as the speed of the rotating member is reduced, the vehicle is brought to a stop—unless the wheel is caused to slide on the road.

Up to about 1920 all cars built in the United States had braking force applied to the rear wheels only, but since that time it has become the general practice to fit brakes acting on all four wheels, which makes it possible to stop a car in practically half the distance, from a given speed. The rotating part of the brake may be applied either directly to a road wheel or to a transmission member in permanent driving relation with the road wheel.

Tread and Wheelbase—The distance between the center lines of ground contact of wheels on opposite sides of the vehicle is known as the track or tread. The National Automobile Chamber of Commerce has adopted a standard of 56 in. for the tread of passenger cars, but there is no standard tread for trucks and buses. The lighter types of both classes of vehicle adhere rather closely to the 56-in. passenger car standard, but in the heavier types treads increase to as much as 72 in., and in the street car type of bus even to 80 in.

The adoption of a tread of 56 in. in the early years of the industry was due to the fact that all horse vehicles in the Northern part of the country had a tread of 56½ in., measured from outside to outside of tires, hence an automobile with 56-in. tread would run in ruts made by horse vehicles. In the South, horse vehicles were made with a tread of 60 in., and during the past some automobile manufacturers have built cars with a 60-in. tread for that section of the country. But as the proportion of hard-surfaced roads increases the need for a standard tread diminishes. When the low-pressure pneumatic tire was introduced, about 1924, owing to its greater width as compared with the high-pressure tire, especially on large passenger cars, it became impossible to provide a rear seat wide enough to be comfortable for three normal-sized passengers, for which reason a 58-in. rear tread is now widely used. In commercial vehicles the front and rear treads are seldom the same. Usually the rear tread is wider, being determined by the width of the body, but sometimes the front tread is made wider, to permit of a greater steering lock, and a shorter turning radius.

In Central and Western Europe, where conditions of rutted roads never had to be considered by designers, the majority of passenger cars have treads considerably smaller than the American standard, the smallest used being 40 in. With these cars, of course, the designers do not attempt to provide seats for three passengers side by side.

The wheelbase of a car varies with the weight and power of the car and also with the type of engine. It is natural that with an engine having eight cylinders in line, a larger wheelbase is required than with a four-cylinder engine. A long wheelbase is of advantage in that it adds to the stability of the car at speed, but it is a disadvantage in that it calls for more storage and parking space, and makes the car harder to manoeuvre.

In the United States only small passenger cars are equipped with four-cylinder engines, the displacements of these engines not exceeding 200 cu. in., and the wheelbases of these cars range from 100 to 110 in. Wheelbases of passenger cars with six-cylinder and eight-cylinder vertical engines average about as follows:

Displacement, cu. in.	Wheelbase, in.	Displacement, cu. in.	Wheelbase, in.
150	105	240	120
180	110	270	125
210	115	300	130

If the same principles of design were rigidly followed the eight-cylinder chassis would come out longer than the six, but in many cases eight-cylinder engines have been designed to replace six-cylinder types without changing the chassis. There is, moreover, a tendency to lengthen the wheelbases of six-cylinder cars beyond what is actually required, as the public is inclined to judge the size of a car by its length of wheelbase. An eight-cylinder car naturally calls for a long wheelbase and the same tendency therefore does not exist.

A V engine is shorter than an "in-line" engine of equal displacement and requires less length on the chassis, thus permitting of reducing the length of wheelbase.

For a given length of wheelbase European cars carry engines of much smaller displacement than cars of American production. This is due to the fact that the annual tax, which is figured on the basis of either the displacement or the cylinder bore, is much higher in Europe than here and compels the use of small engines in cars intended for large production. Insurance rates also are based on engine displacement and further influence designers to keep displacements down.

The wheelbases of buses depend upon the number of passengers which can be seated, upon the comfort of the seating accommodations, and upon the efficiency of utilization of chassis space. More space must be provided per passenger in buses for long distance travel than in city buses, and in

the street-car type the floor area is utilized more efficiently than in the conventional type. Thus the wheelbase for a bus of given passenger capacity depends upon whether the bus is intended for city or inter-city use, whether it is of the conventional or the street-car type, and also, of course, whether it is a single-deck or a double-deck bus, though the latter are built only in capacities that are practically beyond the limit for the former. For conventional single-deck buses for urban and interurban service the relations between seating capacity and wheelbase average as follows:

Seating Capacity	Length of Wheelbase (In.) Urban	Interurban
16	140	165
20	162	187
24	184	208
28	206	230
32	228	
36	245	

The tread of buses varies with the wheelbase as follows, on the average:

Wheelbase (in.)	150	180	210	240
Tread (in.)	57	60	66	75

Motor trucks in capacities of 1 ton and up are generally made in different lengths of wheelbase, to suit the particular service for which they are intended, but each maker has a standard wheelbase for each size of truck, and for four-wheel trucks with two-wheel drive these average as follows:

Capacity	Wheelbase	Capacity	Wheelbase
1,000 lbs.	110 in.	3 tons	163 in.
1,500 lbs.	118 in.	4 tons	173 in.
1 ton	131 in.	5 tons	182 in.
2 tons	150 in.		

Road tractors for hauling semi trailers are built in wheelbase lengths of 125–133 in. for capacities of from 5 to 15 tons.

Wheelbases of farm tractors average as follows:

Capacity	2-plow	3-plow	4-plow	5-plow
Wheelbase	76 in.	83 in.	90 in.	98 in.

CHAPTER II

Friction Clutches

In a motor vehicle propelled by an internal combustion engine and in which the power is transmitted to the wheels by mechanical means, it is necessary—because the engine is incapable of starting under load—to incorporate a friction clutch in the transmission line. The provision of such a clutch makes it possible to start the engine while disconnected from the driving mechanism, and then, by engaging the clutch gradually, to slowly bring the vehicle up to speed.

The friction clutch, moreover, serves also to disconnect the engine from the transmission when it is desired to change gear. With the sliding type of change gear, now in almost universal use, the clutch is normally held in engagement by a spring or springs; when the gears are to be changed it is disengaged by pressing on a pedal, and after the gears have been shifted it is allowed to engage again by releasing the pedal.

Single Plate Clutches—There has been quite an evolution in clutch types since production of motor vehicles first began, the type now in most extensive use being the so-called single-plate, dry type. The clutch is generally combined with the flywheel, which forms one of its members. A typical single-plate clutch is shown in section in Fig. 1. The flywheel web A and the pressure plate B form the driving members, and the single plate C, which is clamped between them, the driven member. Plate C has facings of friction material riveted to it on both sides, and its drop-forged center or hub is mounted on the splined portion of the clutch shaft H. The coiled spring S, seated against the outside of the clutch cover plate D, presses against a flange on sleeve E. From this sleeve the spring pressure is transmitted to the pressure plate B and at the same time multiplied by means of a set of three two-armed levers F. These levers have their fulcra on the inside of the clutch cover plate, and bear against cam surfaces formed on the pressure plate. The cam surfaces form

inclined planes, and by moving the cover plate angularly relative to the pressure plate, the clutch can be adjusted to compensate for wear of the friction linings. A suitable mechanism is provided for locking the cover plate in position after this adjustment has been made.

In order to disengage the clutch, pressure is exerted on a pedal P which is mounted on shaft G. This shaft also

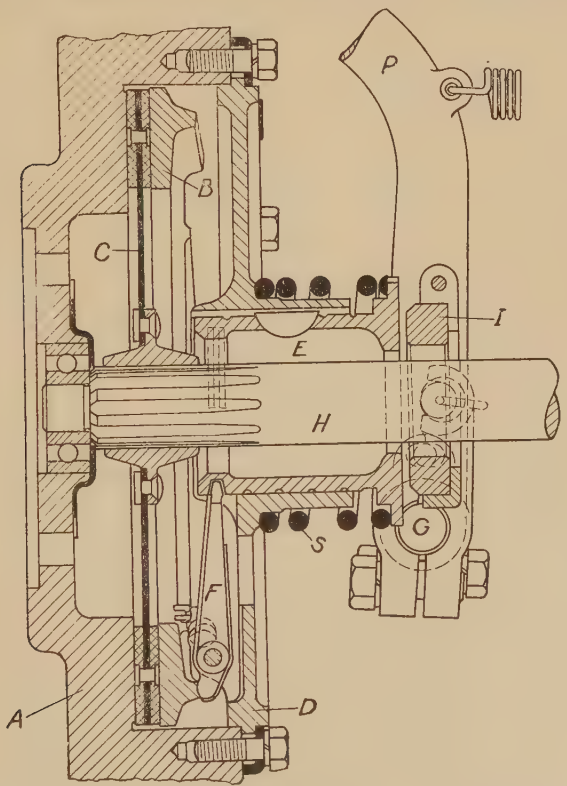


Fig. 1.—Cross-Section of Typical Single-Disc Clutch (Borg & Beck).

carries a throwout yoke which presses on thrust bearing I and through it on sleeve E, forcing the sleeve to the left and thereby withdrawing the pressure plate B from the driven plate C.

Single-plate clutches are made in a number of different types. The type here described is known as the single-spring, "push-in" type, the latter because the release collar must

be pushed inward in order to release it. Single plate clutches are also made in which the plates are pressed together by means of a series of coiled springs, pressing directly against the pressure plate. In that case the number of springs is generally a multiple of the number of levers.

There are several advantages in using the flywheel as one member of the clutch. Cast iron, of which it is made, is the best material for use with asbestos-base friction lining, as its graphite particles seem to have a lubricating effect. High-carbon steel is the next best material, while soft steel is less satisfactory, on account of its tendency to score. Another advantage is the great heat absorbing capacity of the flywheel, which takes care of any abnormal generation of heat, provided it does not continue too long. The second driving member, which is generally also of cast iron, may be regarded as part of the flywheel, as it rotates with the latter at all times and helps it to smoothen out the torque curve.

Friction Facings—Two kinds of friction facing are used for clutches, both having a base of asbestos fiber. One is the so-called molded facing, while the other is woven. Molded facings are punched from rolled plates of short fiber asbestos mixed with a small proportion of binder. These discs are impregnated with asphaltic material and cured for several hours at a high temperature, and after cooling they are straightened, drilled for rivet holes and ground all over, that is, on both their sides and their edges. The woven friction material is made of cords having a brass- or copper-wire core, around which long-fiber asbestos and cotton is spun. This facing, after being woven, is formed into annular disks which are compressed by hydraulic pressure and impregnated and finished by grinding in much the same way as the molded discs. The ends of a section of which an annular disc is formed are joined together by metal staples.

Molded friction discs are dense and tenacious, and, owing to the fact that they have no cotton or appreciable quantities of other heat-sensitive material in them, will withstand comparatively high working temperatures, whereas woven discs must not be allowed to reach such a temperature that the cotton will char. An average composition of a woven asbestos friction fabric is 50 per cent asbestos, 25 per cent wire, 10 per cent cotton and the remainder impregnating and other materials. The woven disc, which is somewhat more expensive to manufacture, has the advantage over the molded disc that it possesses a higher degree of elastic compressibility, which is conducive to smooth engagement and is essential particularly in single-plate clutches, unless the element of

yieldability is provided by the metallic members of the clutch. A friction coefficient of about 0.28 can be figured on with both materials. Clutch facings are ordinarily made in thicknesses of 1/8 in. and 5/32 in.

Calculation of Disc Clutches—Let P be the total pressure exerted by the spring or springs of the clutch and r the ratio of multiplication of any levers by which the spring pressure may be transmitted to the discs. The product Pr is the pressure tending to press the discs together, but not all of this is effective at the friction surfaces. The discs naturally encounter some frictional resistance where they

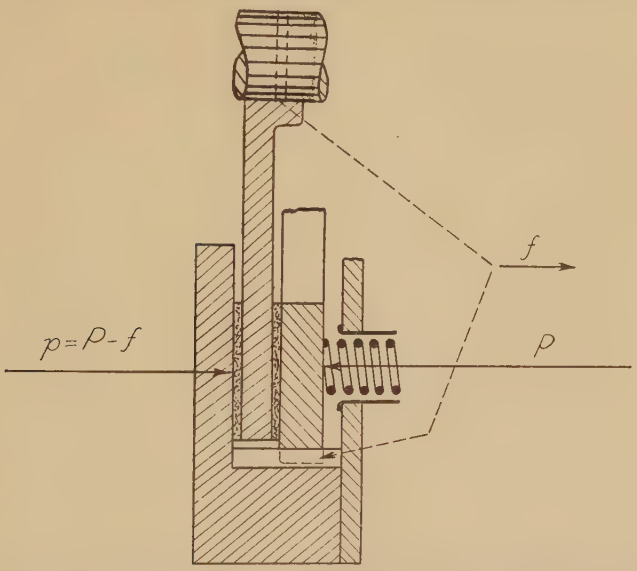


FIG. 2.—DIAGRAM ILLUSTRATING EQUILIBRIUM OF FORCES WHEN CLUTCH IS ENGAGED.

slide on keys, splines or pins, and this resistance must be subtracted from the total pressure. As long as there is no driving action, the resistance to the lateral motion of the discs is slight, but as soon as they come in contact the transmission of torque begins, which means that the pressure at the sliding surfaces on the keys, splines or pins also increases, and with it the resistance to the sliding motion. In clutches with lever multiplication there is, moreover, a certain frictional resistance at the fulcra and points of contact of the levers. All friction along lines in which motion takes place when the discs are pressed into contact must be subtracted

from the gross pressure, and inasmuch as the various surfaces are not lubricated, this is of considerable amount. It can be allowed for by introducing a coefficient of mechanical efficiency e, which in single- and double-plate clutches has a value of about 0.8 and in clutches with 11 or more discs drops as low as 0.50. Fig. 2 illustrates the equilibrium of forces in the direction normal to the clutch discs.

The same as in mechanical structures, we have to allow a factor of safety, which, in this case however is more appropriately called a factor of reserve. If the clutch were designed to carry exactly the maximum power of the engine when new, then the least wear on its friction facings or the least deterioration of same from the frictional standpoint would cause its capacity to drop below that of maximum engine power, and would cause slipping to occur whenever the throttle was opened wide. The factor of reserve naturally should be somewhat greater in clutches with no provision for adjustment. Where means of adjustment are provided it may be taken at 1.25 and in the opposite case, at 1.50.

The number of effective friction surfaces is equal to the sum of the driving and driven discs minus one. In this number have to be included the end discs, usually constituted by the flywheel web and the pressure plate. Thus in a so-called single plate clutch, in which there are one driven and two driving discs (flywheel web and pressure plate) there are $3 - 1 = 2$ friction surfaces. There is one type of disc clutch (now rarely used) in which the friction discs float, being keyed neither to the driving nor to the driven member, and these floating members are not counted.

Let the outside diameter of the friction surfaces be D and the inside d, then the area of each friction surface is $\pi/4\ (D^2 - d^2)$; and if the gross specific pressure on the friction surfaces is p lbs. per sq. in., the total pressure on each surface is $\pi/4\ (D^2 - d^2)\ p$. The net specific pressure on each surface is found by multiplying by the mechanical efficiency e:

$$P_n = \pi/4\ (D^2 - d^2)\ p\ e,$$

and the friction produced by this pressure between discs is

$$F = \pi/4\ (D^2 - d^2)\ p\ e\ f,$$

where f is the coefficient of friction.

The moment of this frictional force around the axis of the clutch is

$$\begin{aligned} M &= \pi/4\ (D^2 - d^2)\ p\ e\ f\ (D + d)\ /4 \text{ lb.-in.} \\ &= \pi/4\ (D^2 - d^2)\ p\ e\ f\ (D + d)\ /48 \text{ lb.-ft.} \end{aligned}$$

Then, if n is the sum of the driving and driven discs, there are $n - 1$ effective friction surfaces and the total moment of the clutch, or its limiting torque capacity, is

$$M_l = \pi/4\ (D^2 - d^2)\ p\ e\ f\ (D + d)\ (n - 1)\ /48 \text{ lb.-ft.}$$

Allowing a factor of reserve R the rated capacity becomes,

$$M_r = .\pi/4\ (D^2 - d^2)\ (1/R)\ p\ e\ f\ (D + d)\ (n - 1)\ /48 \text{ lb.-ft.}$$

The value of f is to be taken at 0.28; in a single-plate clutch for passenger-car use p will vary between 25 and 35 lbs. per sq. in. and has an average value of 28 lbs., while the mechanical efficiency e may be taken at 0.8 and the coefficient of reserve R at 1.25. The value of n is 3.

A clutch of this type of 9 in. outside diameter of friction surfaces and 5.5 in. inside diameter therefore should have the following rated capacity:

$$(3.14/4)\ (9^2 - 5.5^2)\ (1/1.25) \times 28 \times 0.8 \times 0.28 \times (9 + 5.5)\ (3 - 1)/48 = 136 \text{ lb.-ft.}$$

In the case of clutches of this type for use on buses, where the vehicle must be stopped and restarted continuously, it is customary to reduce the gross specific pressure to about 18 lbs. per sq. in. This same specific pressure is used in multiple disc clutches, in which it is an easy matter to get sufficient surface.

Allowable Unit Pressure Varies with Diameter—If both the unit pressure and the speed of revolution were constant, the heat generated in the clutch would vary as the cube of its mean diameter, whereas the radiating surface varies only about as the square of that dimension. As the heat generated must not increase any faster than the radiating surface, it follows that the unit pressure and the speed cannot both remain the same when the mean diameter is increased. In fact, their product should vary inversely as the mean diameter.

Engine speed naturally decreases somewhat with increase in size, and it is often assumed to be inversely proportional to the square root of the piston stroke or of the linear dimensions of the engine cylinders. This would make it inversely proportional also to the mean diameter of the clutch friction surfaces. Therefore, if the unit pressure on the friction surfaces also were made inversely proportional to the square root of the mean diameter, the heat generation would vary as

$$D^3 \times \frac{1}{\sqrt{D}} \times \frac{1}{\sqrt{D}} = D^2.$$

or the same as the radiating surface. Hence we may say that for similar conditions (as regards frequency of clutch operation) and with similar design (particularly as regards ventilation), the unit pressure on the friction surfaces should vary inversely as the square root of their mean diameter.

Permissible Pedal Pressure—In a passenger car the pressure required on the pedal in order to disengage the clutch should not be less than 20 lbs. and not more than 30 lbs. The reason for the lower limit is that many drivers are in the habit of resting their foot on the clutch pedal while driving, and if the pressure required to disengage is quite small, they are likely to cause the clutch to slip continually. This is known as "riding the clutch." Such slippage, of course, entails needless power loss and heating. The reason for the upper limit is that the clutch must not be too hard to operate,

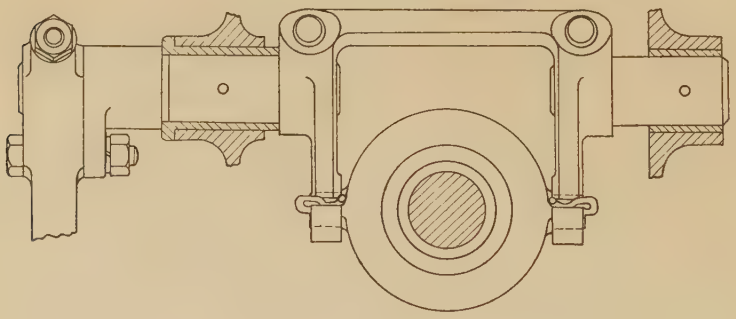


FIG. 3.—ONE FORM OF CLUTCH YOKE.

especially in view of the fact that a great many passenger cars are now being driven by women. A good average pressure is 25 lbs. In motor trucks a pedal pressure of 35–40 lbs. is permissible.

Clutch pedals vary in length from 14 to 18 in., while the yoke arms on the pedal shaft which press against the clutch throw-out collar measure from 1¼ to 2½ in. in length. Pressure multiplication ratios of from about 5½ to about 14½ are thus obtained. The multiplication ratio must be made equal to the quotient obtained by dividing the pressure required on the throw-out collar by the allowable pressure on the pedal. For instance, if a pressure of 300 lbs. is required on the throw-out collar and it is desired to limit the required pedal pressure to 25 lbs., the pedal leverage must be 300/25 = 12.

In order to make it possible to reduce the length of the

short lever arms to the dimensions given, it is necessary to make them in the form of a yoke, with the supporting trunnions set in toward the free ends of the levers. As shown in Fig. 3, the yoke proper is generally made as a forging, into which the supporting trunnions are fastened.

If it is impossible to get the desired leverage with a simple lever, compound levers, such as illustrated in Fig. 4 are made use of. Naturally, the use of compound levers is more common in powerful cars and trucks.

Moment of Inertia—A consideration of prime importance in the design of a clutch for use with a sliding pinion change gear is that the moment of inertia of the driven part must be as small as possible, as to it is directly proportional the shock imparted to the gear teeth when the gears are brought into mesh while their pitch-line velocities are unequal. It is this which has led to the wide use of single-plate clutches, the driven part of which consists of a single disc of sheet steel of from $\frac{3}{32}$ to $\frac{1}{8}$ in. thickness.

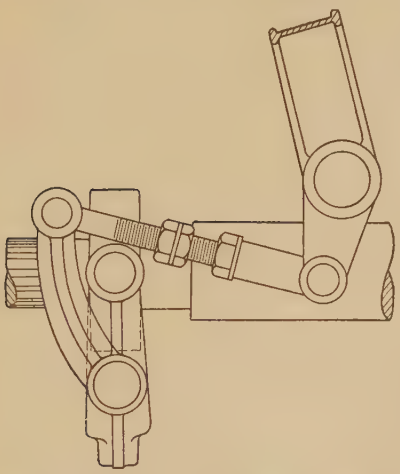


FIG. 4.—COMPOUND CLUTCH LEVERS.

Generally, the driven disc is faced with two annular discs of friction material. In some cases, in an endeavor to further reduce the inertia of the driven part, designers have secured the friction linings to the driving members. This has the disadvantage that the frictional heat is then generated between the friction discs and the driven member and must be carried away by the latter, which has much less heat dissipating capacity than the driving members—the flywheel and the pressure plate. Undoubtedly the moment of inertia can be reduced in this way, and the heat dissipating requirements can be met by increasing the size and by operating the clutch in oil or by providing ventilation.

Limit of Capacity—It may here be pointed out that the capacity of any given design of clutch depends essentially upon its ability to get rid of the heat generated. All friction linings have a critical temperature, at which the binding or

impregnating materials "fry out," causing the lining to lose its frictional characteristics. This temperature must not be attained in practice. The mere ability to pull a torque load can be readily increased by increasing the spring pressure, but overheating sets a definite limit to the capacity. This limit, of course, can be increased by increasing the surface area of the parts on which the heat is generated, and by providing air circulation through the clutch.

Another plan that permits of reducing the moment of inertia of the driven member, and which has been used particularly in multiple disc clutches, consists in letting the friction discs float, that is, attaching them neither to the driven nor to the driving metal discs. An objection to this plan is that in order to prevent dragging, the throw-out collar must be moved a greater distance for a given total number of driving and driven members, making clutch operation harder.

Means to Ensure Smooth Engagement—Generally speaking, friction clutches are liable to three kinds of faulty operation. In the first place, they may slip, owing either to insufficient spring pressure or to the presence of lubricant on the friction surfaces. The remedy for either condition is obvious. Secondly, they may grab; that is, engage too abruptly, which is generally due to too high spring pressure and excessive capacity of the clutch for the work to be done. Thirdly, the clutch may shatter.

Shattering is most likely to occur with molded friction lining, owing to the dense, unyielding character of the latter, and where this material is to be employed it is practically necessary to so design the metallic parts of the clutch that they can yield slightly as the clutch is engaged. It has been found that absolute uniformity of thickness of the facings tends to prevent shatter.

One method of providing yieldability is that employed in the Durant clutch, the driven member of which is illustrated in Fig. 5. The outer portion of the driven disc is provided with radial corrugations, and segmental sections of friction materials are riveted to both sides of the disc, covering a little more than half of the surface on each side. The center of each segment coincides with the crest of the corrugation of the disc. As pressure is applied to the discs, the corrugations flatten out gradually, thus ensuring a gradual increase in the friction and in the driving torque, and smooth engagement.

Another example of a flexible driven member is afforded by the Long clutch, the driven member of which is shown in

Fig. 6. This member is spoked, and its outer portion is dished slightly, so that the inner and outer edges of the friction lining come into contact with the driving members first, the driven member flattening out as more pressure is applied. Driven discs designed to possess yieldability are made of saw steel. In the case of the Long clutch the driven member has its outer portion formed into a 3 deg. cone while in the annealed state and is then tempered. Figs. 5 and 6 are repro-

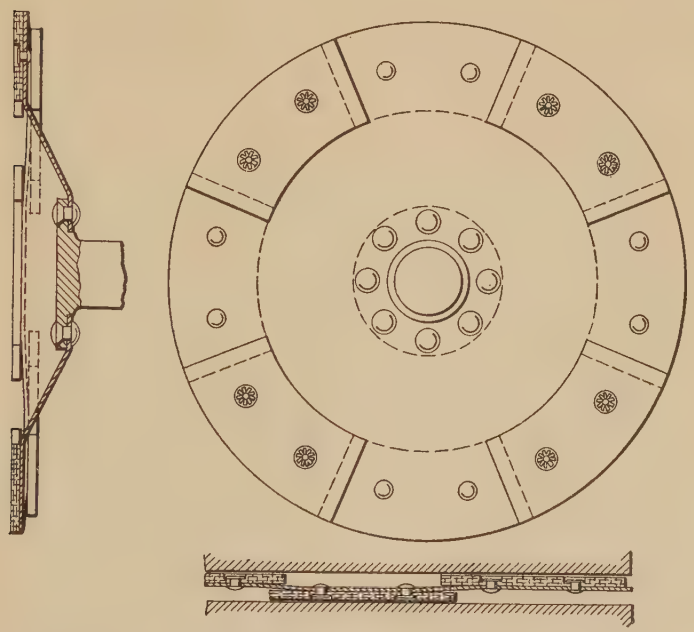


Fig. 5.—Clutch Driven-Member with Sectors of Friction Material Alternating on Opposite Sides (Durant).

duced from a paper read before the Society of Automotive Engineers (S. A. E.) by E. E. Wemp.

Flexible Levers—Still another method of ensuring yieldability and smooth engagement is embodied in the Brown-Lipe single-plate clutch used on the Franklin car, as shown in section in Fig. 7. The pressure of the single central spring is multiplied and transmitted to the pressure plate by means of 40 flat-bar-type levers. With the type of mechanism employed in this clutch, the pressure plate is not positively withdrawn when the clutch is disengaged, and for this reason

a number of small coiled springs are provided near its periphery, and anchored to posts secured into the flywheel rim.

Adjustable and Non-adjustable Clutches—The Borg & Beck and the Brown-Lipe clutches, illustrated in Figs. 1 and 7, respectively, are provided with means of adjustment to permit of compensating for the wear of the friction facings. In the former the multiplying levers bear on cam surfaces on the pressure plate, and the clutch cover plate, on which the levers are fulcrumed, is so arranged that it can be shifted angularly. After the clutch facing has worn a certain amount, the cover plate is released and turned slightly in a

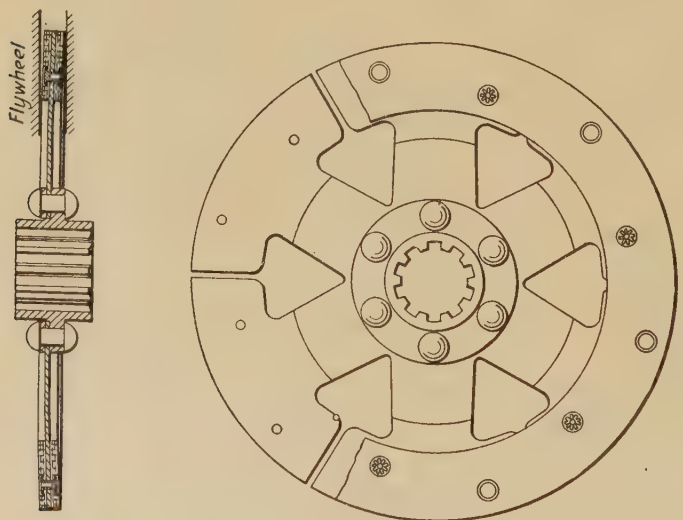


FIG. 6.—DISHED, SPOKED DRIVEN MEMBER (LONG).

right-handed direction, looked at from the rear end, the short end of the lever then taking purchase on a higher spot of the cam surface, thereby restoring the original pedal action.

In the Brown-Lipe clutch the outer ends of the flat levers pressing against the pressure plate bear against a threaded collar screwed into a threaded insert in the flywheel. By screwing this collar further out of the insert, wear of the friction facings can be compensated for. The non-adjustable type of clutch has been made practical by the development of a dense molded clutch facing which wears very slowly.

In some clutches, in addition to the multiple levers, wedges or toggle mechanisms are employed to multiply the

pressure on the clutch pedal and possibly also that of the springs. With a toggle mechanism the ratio of pressure multi-

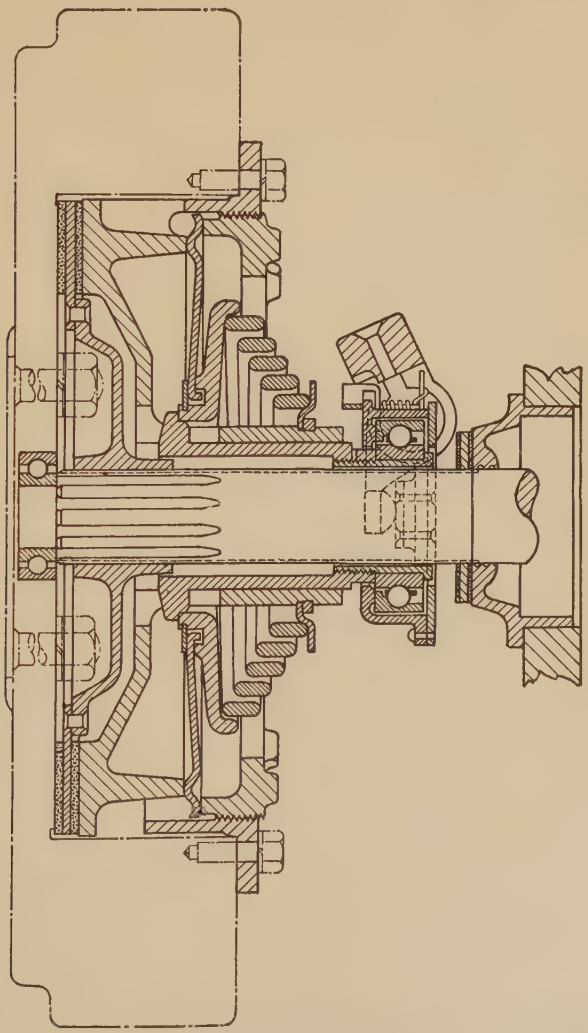


FIG. 7.—CLUTCH IN WHICH SMOOTH ENGAGEMENT IS SECURED BY FLEXIBLE LEVERS.

plication is variable with the position of the throw-out sleeve, being greatest when the clutch is in; but such mechanisms

call for very close adjustment. In the conventional single-plate clutch, either three or four levers are used. If it is impossible to get the required leverage when the levers are radial, they may be placed in a tangential direction. The clutch as a whole must be perfectly balanced and the design of the mechanism must therefore be symmetrical.

Asbestos-faced disc clutches can be operated also in oil. According to E. E. Wemp (S. A. E. Journal, Oct., 1925), if the usual half-and-half mixture of cylinder oil and kerosene is used, the spring pressure must be increased 30 per cent.

Clutch Bearings—In some of the lower-priced cars the need for a thrust bearing or its equivalent is obviated by providing the arms of the clutch throw-out yoke with rollers which bear against a radial flange on the throw-out sleeve when the clutch is in the disengaged position, and are held free of the flange by a spring when it is engaged. (Fig. 8). The objection to this construction is that there is sliding friction between the rollers and their studs; owing to their inaccessibility, the lubrication of the rollers is likely to be neglected, and they are then apt to squeak and even to seize.

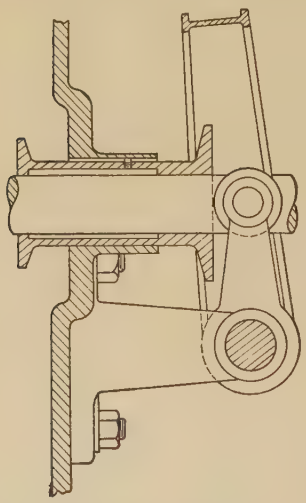


FIG. 8.—THROW-OUT YOKE WITH ROLLERS BEARING AGAINST SLEEVE-FLANGE.

A second method, illustrated at the left in Fig. 9, consists in the use of an annular block of graphite which transmits the pressure of the yoke from a trunnion plate to a flange on the clutch sleeve.

Three types of ball bearing are used for the clutch throw-out. One of these is the regular annular ball bearing, which, preferably, should be of the deep groove type, on account of the greater capacity of the latter for thrust loads. The second is the radial-axial type of bearing, in which the line of contact is inclined to the axis of rotation; and the third, the regular ball thrust bearing.

The annular and radial-axial types are preferable, because they will carry radial as well as thrust loads, and it is difficult to entirely avoid radial loads on these throw-out bearings. If there is any appreciable radial load on the plain, grooveless type of thrust bearing, there will be sliding contact at one of the mountings, and a groove may be worn in it. The an-

nular and radial-axial types of bearing are more expensive, however, and are used only on the higher-priced vehicles and tractors.

The bearings should always be well housed, so they will retain the grease in which they are packed, and so that dust and dirt will be kept out. A mounting for an annular type of bearing is illustrated at the center in Fig. 9. The inner race is clamped between a split collar and a nut on the clutch sleeve, with thin washers between, while the outer race is clamped in a two-part housing, one part of which is formed with a flange, against a machined surface on which the yoke arms bear.

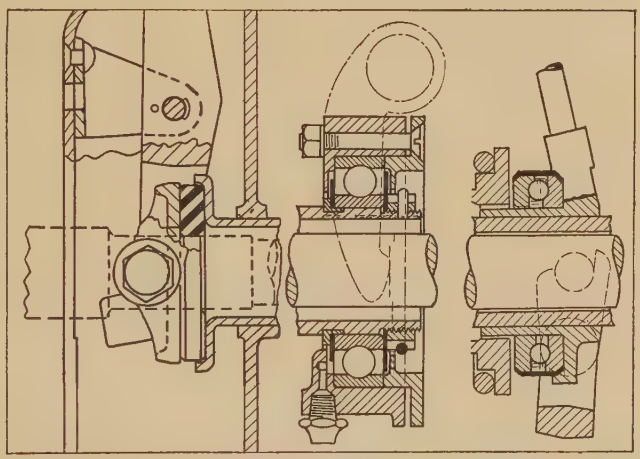


FIG. 9.—THREE TYPES OF CLUTCH THROW-OUT BEARING AND THEIR MOUNTINGS.

A mounting for a ball thrust collar is shown at the right in Fig. 9. The bearing is mounted on the clutch sleeve against a thrust flange thereon, which is provided with trunnions for the throw-out yoke and a boss for an oil tube. This sleeve rides on a tube on the transmission bearing cap. When the clutch is disengaged, one race of the thrust bearing is pressed against a flange at the rear end of the regular clutch sleeve.

Clutch Throw-out Sleeve—When the clutch is engaged or disengaged the throw-out collar and thrust bearing slide in an axial direction, and in order to properly guide them in this motion they are mounted on a sleeve. This sleeve may be mounted or supported in four different ways: (1) on the clutch shaft; (2) inside a bearing formed on the clutch cover

plate; (3) on the outside of a hub formed on the clutch cover plate; (4) on the outside of a hub or sleeve formed on the transmission bearing cap.

In judging the respective merits of these various mountings it must be borne in mind that it is desirable that there should be no sliding motion between parts when the clutch is in engagement, and it is also desirable that both races of any anti-friction bearing used in the mechanism should not rotate under this condition. The clutch throw-out bearing and the sleeve on which it is mounted are particularly difficult to lubricate effectively, for one reason because they are rather inaccessible, and for another because, if the bearing rotates when the clutch is engaged, centrifugal force tends to cause the lubricant to be thrown off.

When the clutch sleeve is mounted on the transmission bearing cap, it and one of the races of the ball bearing are non-rotating at all times; when they ride on the clutch shaft they rotate as long as power is being transmitted, whereas when they are supported by the clutch cover plate, they rotate whenever the engine is running. When the throw-out bearing rides on the clutch shaft, the clutch pedal is provided with a retracting spring and is so adjusted that when the clutch is engaged there is a clearance of 1/16 in. or so between the throw-out collar and the yoke arms, so that the throw-out bearing will not be working when the clutch is in.

Clutch Lubrication—The problem of lubrication arises chiefly with the dry type of clutch, for if the interior of the clutch is partly filled with lubricant it is not difficult to get sufficient to the bearing surfaces.

In dry disc clutches lubricant must be supplied to the clutch pilot bearing and to the clutch throw-out sleeve and its bearings (unless the latter happens to be of the self-lubricating type). The problem is complicated by the requirement that none of the lubricant supplied to these bearings must get into the clutch proper, as it would cause the friction members to lose some of their holding power.

The clutch pilot bearing, which is generally either a Hyatt roller bearing if mounted in the end of the crankshaft, or a small size ball bearing if in the bore of the flywheel web, may be lubricated from the engine lubricating system by drilling a radial hole in the crankshaft rear bearing, placing the bearing mounting space in communication with a collector groove in the rear bearing. A method of guarding against the escape of oil from this bearing into the clutch is illustrated in Fig. 10.

Where the clutch throw-out collar is not enclosed in a

flywheel housing it is merely provided with a lubricator fitting, so that grease can be forced into the ball bearing and to the bearing surface of the sleeve when required. If it is enclosed it is at least desirable to bring a connection from the collar to the outside of the housing, and since the collar must slide, this connection must either be flexible or a sliding fit in a slot in the wall of the housing. Owing to the importance of this bearing, a flexible tube has sometimes been carried from it to the outside of the car frame and there provided with a lubricating fitting or screw-down grease cup.

Another plan consists in drilling through the axis of the clutch shaft from the transmission end and lubricating the clutch bearings from the transmission. In the Durant clutch the throw-out sleeve is provided with a bulb of pressed steel which can be filled with grease.

Clutch Springs—The springs which hold the clutch in engagement are generally of the coiled type, although leaf springs have been used in some instances. The formulae for the maximum safe load and compression under load of round-wire springs are as follows:

$$W = \frac{\pi S d^3}{8E} \qquad F = \frac{8nPD^3}{Ed^4}.$$

For springs made of wire of square section with length of side d the formulæ become

$$W = 0.471\,\frac{Sd^3}{D} \qquad F = 4.71\,\frac{nPD^3}{Ed^4},$$

where

D = mean diameter of the coil in inches;
W = maximum safe load in pounds;
F = compression of the spring;
d = diameter or length of side of the wire, in inches;
n = number of coils in the spring;
S = maximum safe fiber stress for the material;
E = torsional modulus of elasticity (12,500,000);
P = load in pounds.

In clutches for unit power plants it is essential that the springs be as compact as possible, and for this reason springs of square or rectangular section wire are often used, which for the same over-all dimensions will carry a larger maximum safe load. These springs are sometimes set into counterbored holes in the flywheel rim (a practice in favor in England) or in pressed steel thimbles with flanges at their open ends, which are set into the clutch-cover plate.

In order to still further reduce the axial length of the space necessary for the spring, volute-type springs, as shown in Fig. 11, are sometimes used. These, too, may be either of round or rectangular section wire.

Clutch Shaft—The shaft which carries the driven member is known as the clutch shaft, and in unit power plants this same shaft is also the primary shaft of the transmission, extending into the transmission housing and having certain parts of the transmission formed integral with it. In unit

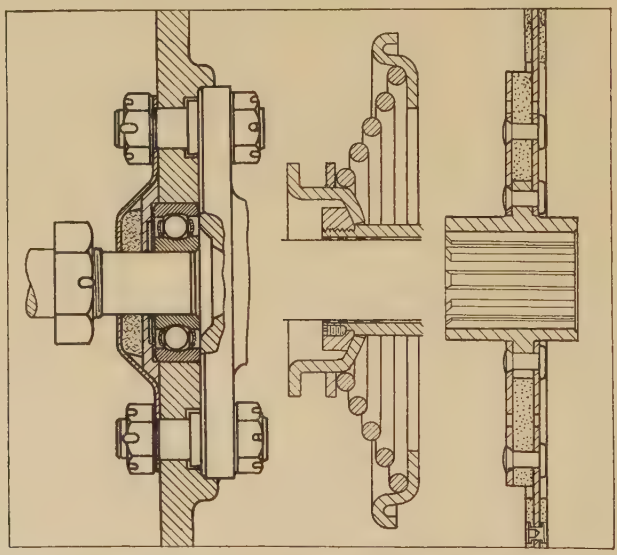


Fig. 10 (left).—Oil-Guard for Clutch-Pilot Bearing.
Fig. 11 (center).—Clutch Spring Designed for Axial Compactness.
Fig. 13 (right).—Driven Member with Flexible Insert.

power plants this shaft is generally supported in a pilot bearing in the flywheel and in a bearing in the forward wall of the transmission housing. In cases where the transmission is a separate unit the clutch shaft may be supported in bearings in the flywheel and in the clutch cover plate, or solely in a long bearing in the clutch cover plate. In a bearing of the latter type the shaft rotates in its bearing only when the clutch is disengaged and the engine is running, and this bearing is therefore of the plain bushed type.

The diameter of the clutch shaft is determined by the

requirement of adequate bearing surface on the splines on which the clutch center is mounted. A ten-spline joint is usually used, of a special design which gives contact only at the sides of the splines. As there is more than enough metal at the bottom of the splines to carry the torque load, the bottom of the grooves is rounded, as this preserves the cutting tools and tends to minimize distortion in heat treatment. The diameter of the shaft over the outside of the splines can be made

$$d = \sqrt[3]{\frac{T}{54}} \text{ in.},$$

where T is the torque capacity of the clutch in lb.-ft. The length of the splined section is made equal to the outside diameter or slightly less.

In the clutch shaft for a unit power plant, which has the constant mesh pinion of the transmission formed integral with it, the diameter of the full-round portion of the shaft is not reduced below that at the top of the splines, as a rule, even though the stress in this portion of the shaft under full torque figures out to only about 5,000 lbs. per sq. in. However, where the transmission is separately mounted and the rear end of the clutch shaft has a permanent splined fitting for one member of a universal joint, the full-round portion of the shaft may be turned down to that at the bottom of the splines of the sliding joint.

Clutch Brakes—With types of clutch having a driven member of considerable moment of inertia, it is customary to provide a clutch brake as an aid in shifting gears without clashing. Smooth shifting of gears can be effected only when the pitch line velocities of the two mating gears are substantially equal, and it is not always easy to secure these equal velocities.

Assume that it is desired to "change down," or from a higher to a lower gear. The first step is to disengage the clutch. The car will continue to run along, by virtue of its inertia, gradually losing speed owing to friction, and for the same reason the driven member of the clutch will keep on running, also gradually losing speed. At the moment of unclutching the speed of the driven gear then engaged was exactly right, while the speed of the driven gear to be engaged was much too high. If the change is made on an up-grade, as is usually the case, the ideal meshing condition is rapidly approached automatically, because the car—and with it the driven gear to be engaged—is rapidly losing speed. The con-

ditions are different, however, when the shift is to be made on the level.

In that case an operation known as double clutching is sometimes made use of. Instead of the driven gear being slowed down, the driving gear is speeded up. When the clutch is first disengaged and the engine is thus freed of its load, it naturally speeds up. Therefore, if the clutch is re-engaged for just an instant, the driven member will be speeded up and caused to approach the ideal meshing speed, thus facilitating the shift.

When a shift is made in the opposite direction, from a lower to a higher gear, after the clutch is disengaged the driven member turns too fast and the car moves too slow

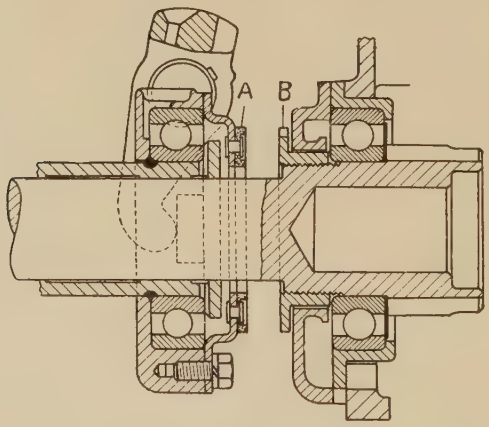


Fig. 12.—Clutch Brake.

for easy shifting. Of course, unless the clutch drags, its driven member will gradually slow down, but the car itself also is losing speed and the conditions of mesh therefore improve little, if any, with time. To make it possible to shift quickly even with a driven member of considerable inertia, a clutch brake is provided. As shown by the example illustrated in Fig. 12, this consists of a member B rotating with the driven member of the clutch, which is adapted to be brought into contact with a stationary member A by the mere act of pressing the clutch pedal all the way down. One of the two engaging members usually has a friction facing. Such clutch brakes (or clutch stops, as they are called in England) are rarely if ever used with single plate clutches, but frequently with the multiple disc type.

Flexible Clutch Center—Clutches for passenger cars are sometimes provided with a driving member having an annulus of rubberized fabric interposed between the frictional rim and the hub (Fig. 13). The reason for this is that multi-cylinder engines are prone to develop torsional vibration in their crankshafts. When this occurs the rotary motion of the flywheel is not uniform, but, on the contrary, its speed fluctuates at a very rapid rate. With a rigid connection between the engine crankshaft and the primary shaft of the

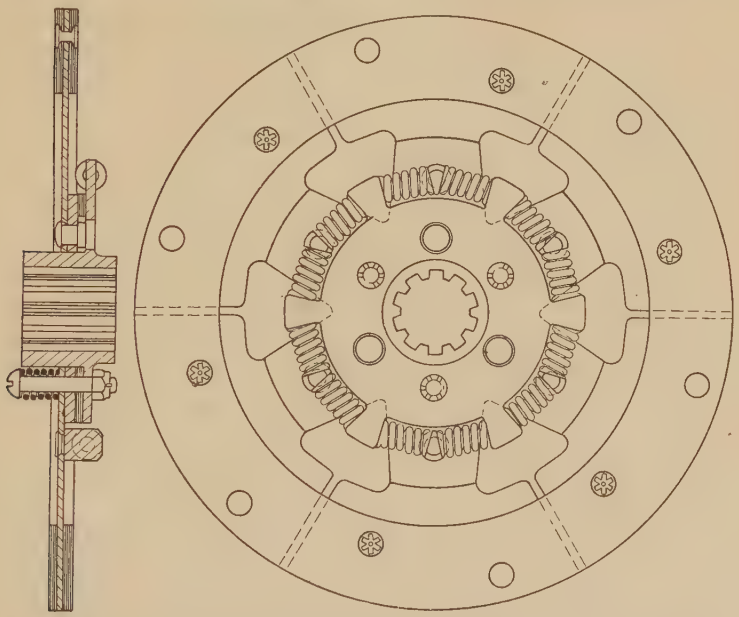


FIG. 14.—LONG CLUTCH WITH CUSHION SPRINGS AND FRICTION DAMPING DEVICE.

transmission, these speed fluctuations are transmitted to the transmission members. The gears of the transmission are always made with a slight clearance between teeth, and the speed fluctuations will cause this clearance to be taken up and gear clatter to occur. With a flexible insert in the clutch driven member these speed fluctuations are absorbed and gear clatter is prevented. Clatter may occur even at the splines of the clutch driven member.

A further development of the vibration-damping idea is represented by the Long clutch, shown in section in Fig. 14.

Torque is transmitted from the driven disc to the clutch center through a series of small coiled springs, which absorb any sudden fluctuations in crankshaft speed. A disc of friction material is riveted to the driven member and pressed against a flange on the clutch center, by means of three bolts with coiled springs under their heads, and when there is any relative angular motion of the disc with respect to the center, there is sliding motion between the friction disc and the

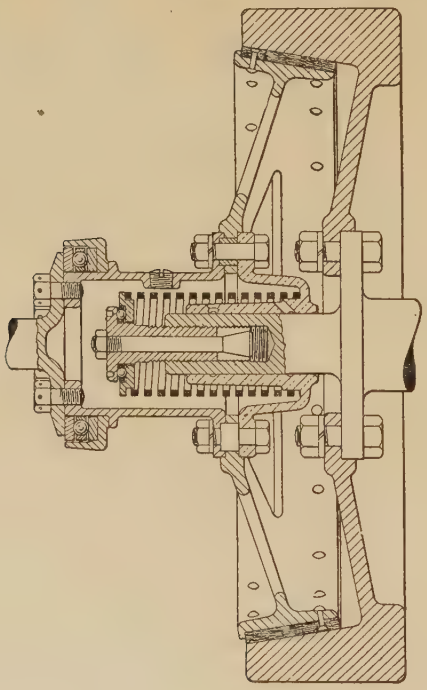


FIG. 15.—CONE CLUTCH, DIRECT TYPE.

flange. This produces a damping action which damps out torsional vibration of the rear end of the crankshaft in the same way as torsional vibration of the front end is damped out by the usual vibration damper of the engine. For proper damping action the pressure of the springs which draw the friction discs together must be accurately adjusted, and for this reason clutches of this type, when their lining has worn down, are not re-lined, but have their driven members replaced.

The hub of the driven member is generally made of a forging of low carbon steel and its splines are case-hardened.

Cone Clutches—Cone clutches may be divided into two classes, the direct cone and the inverted cone, examples of the two types being shown in Figs. 15 and 16, respectively. As shown in Fig. 15, the direct cone is combined with the flywheel, the rim of which is bored out to an angle of about 12°, to form the female cone. Into this the male cone is forced by the pressure of one or more coiled springs. Cone clutches are generally piloted on an extension of the crankshaft. In the inverted cone the female member is formed by

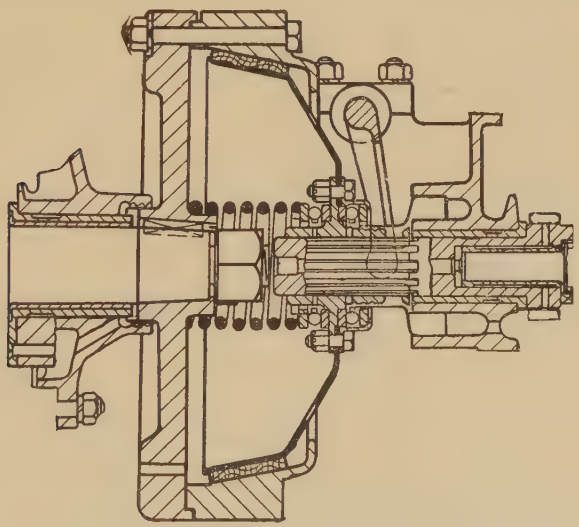


FIG. 16.—CONE CLUTCH, INVERTED TYPE.

a cast iron or steel ring bolted to the rim of the flywheel, into which the male cone enters from the flywheel or engine end. The inverted type of cone is particularly adapted for use in unit powerplants, owing to the fact that it is more compact, the spring being located between the flywheel and clutch cone, instead of to the rear of the latter. Cone clutches are not well adapted for use on engines of high power, and have practically passed out of use here, but are still being used on light cars in Europe. Originally the cones were generally cast in aluminum, but now they are frequently made of pressed steel.

An inverted cone clutch as used on a low-powered British

car with a unit power plant is shown in Fig. 16. The pressed steel clutch cone is bolted to a flange on the clutch center, which latter is a sliding fit on the splined forward end of the transmission primary shaft. A ball thrust bearing is fitted on each side of the flange on the clutch center, that on the forward side taking the thrust of the clutch spring and that on the rear side the thrust of the throw-out yoke. Both bearings carry active or dynamic load only when the clutch is disengaged.

Calculation of Cone Clutches—A diagram of a cone clutch is shown in Fig. 17. The spring pressure P forces the male

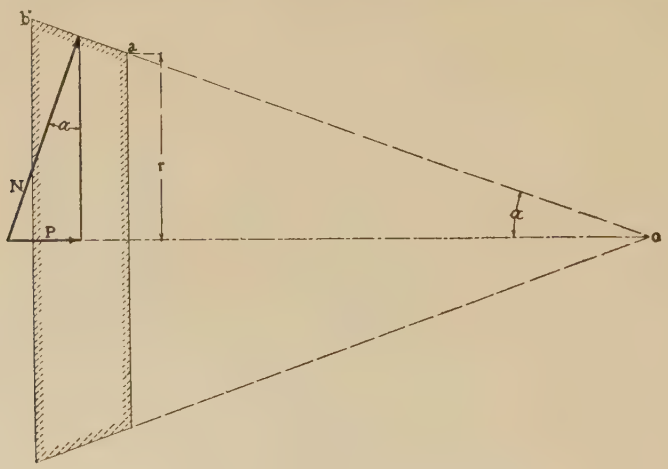


Fig. 17.—Diagram of Cone Clutch.

cone against the female cone, producing a normal pressure N at their contact surfaces. Neglecting the friction between the two surfaces for the moment, according to the principle of the parallelogram of forces,

$$P/N = \sin \alpha;$$

hence

$$P = N \sin \alpha,$$

where α is the angle of the cone. The adherence or frictional force F between the clutch cones is equal to the normal pressure multiplied by the coefficient of friction f,

$$F = Nf = Pf/\sin \alpha.$$

Cone clutches are faced with asbestos fabric, and the same coefficient of friction obtains as in the case of disc clutches.

In cone clutches, however, the same as in disc clutches, there is a certain internal friction which acts to reduce the effective pressure between the contacting surfaces from that calculated from the parallelogram of forces.

When the surfaces first come in contact the pressure between them is light, and little power can be transmitted. They must therefore be pressed into more intimate contact, but to do so involves forcing the surface over each other against the force of friction between them. Referring to Fig. 18, let N represent the effective normal pressure between the frictional surfaces and P the spring pressure necessary to produce this normal pressure. Also, let F represent the frictional force in the direction of a generatrix of the cone, P_1 its component parallel to the clutch axis, and ϕ the so-called friction angle ($\tan \phi = f$), then

$$P = N \sin \alpha,$$
$$F = N \tan \phi,$$

and

$$P_1 = F \cos \alpha = N \tan \phi \cos \alpha.$$

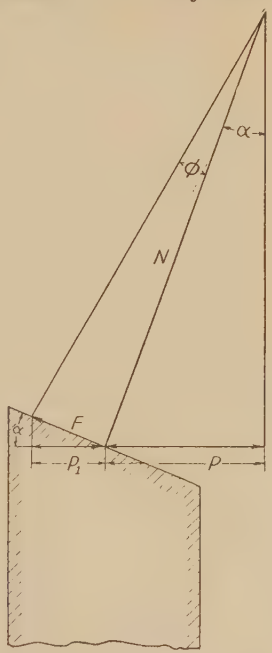


FIG. 18. — DIAGRAM ILLUSTRATING EFFECT OF FRICTION ON EFFECTIVE NORMAL PRESSURE.

The total spring pressure necessary to cause the clutch to engage sufficiently firmly to produce a frictional force N is

$$P_2 = P + P_1 = N (\sin \alpha + \cos \alpha \tan \phi).$$

Metal Disc Clutches—A type that deserves mention as having had an important part in the development of the automobile, even though it is now rarely used, is the metal-to-metal, multiple-disc-in-oil clutch, which displaced the cone clutch on the larger and more expensive cars as engine powers increased beyond what could be readily handled with cone clutches. Most of these clutches had plane discs, but one representative of this type, the Hele Shaw, illustrated in Fig. 19, has alternate discs of steel and bronze with V grooves in them, whose walls form an angle of 35°. Only the walls of the grooves come in contact with each other, and the remainder of the discs serves merely for radiating heat. Oil holes are punched through the inner walls of the grooves near the peak, so the oil can enter and escape freely. It is

obvious that in this clutch there is a sort of wedge action, the same as in a cone clutch, for which reason less spring pressure is required than with a plane disc clutch of the same number and size of discs and transmitting the same power. On the other hand, the discs have to be moved laterally a greater distance in order to ensure proper clearance.

The friction coefficient for lubricated metal discs is naturally much smaller than that for asbestos fabric on dry metal, hence for the same spring pressure a much larger number of discs have to be used than in a dry-disc type of clutch. This results in a great moment of inertia of the driven member, which makes gear shifting difficult. The greatest trouble with the metal-in-oil disc clutch, however, was due to the influence of the weather on the properties of the oil. A half-and-half

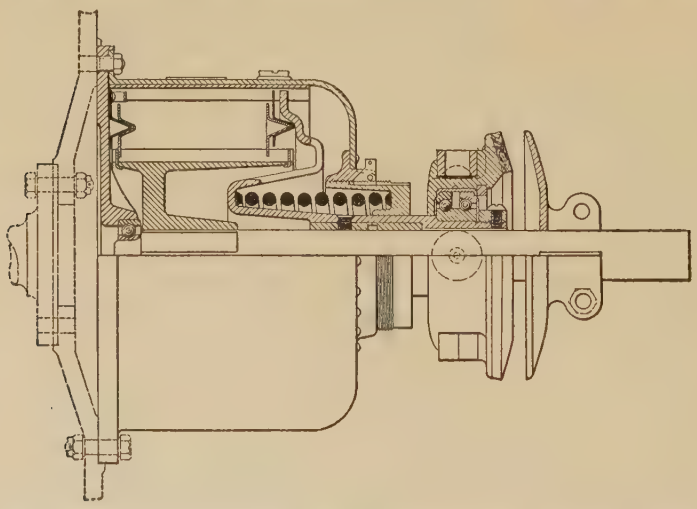


Fig. 19.—Multiple-Disc-in-Oil Clutch with V-Shaped Discs.

mixture of cylinder oil and kerosene was generally used, and in zero weather it sometimes became so viscous that the clutch would drag. The effect of the oil was dependent upon both quantity and quality, and when the oil supply had to be renewed it was difficult to get the right combination.

Multiple Dry Disc Clutches—In order to eliminate trouble from dragging the dry disc clutch was introduced. It comprises two sets of metal discs, one driving and one driven, of which one set, preferably the driven one, is faced with asbestos fabric on both sides. With this type of clutch no levers are used, the spring pressure acting directly in pressing the discs into contact. At one time it was customary to use

a large number of discs and keep the unit pressure quite low, evidently under the impression that this would increase the life of the facings. However, with such a large number of discs there is a tendency for the clutch to drag, and gear shifting also is difficult, because the driven discs naturally have a good deal of weight. Therefore, where the multiple disc clutch is still being used, it usually has a comparatively small number of discs.

Fig. 20 shows a multiple disc clutch of Packard design. There is an outer drum with keys on its inner face on which the driving discs slide, and an inner drum keyed to a tapered seat on the clutch shaft, on which the driven discs slide. The springs are located in the annular space between the outer drum and the flywheel rim, and their pressure is transmitted to the pressure plate by pull rods.

Driving Means—In disc type clutches the discs must be capable of sliding on the parts on which they are supported and still must have a driving connection with these parts. These driving connections may be in the form of studs, keys, gear teeth or bolt heads of a special form acting as keys. The latter type of driving means is used extensively for single disc clutches, in which the only member that has to be driven in this way—the plate rotating with the flywheel—usually is of considerable width at its circumference.

The problem of the driving means assumes considerable importance in a multiple disc clutch. Because the discs are quite thin, if their lugs bear on only a small number of studs or keys, the specific pressure on the contact surfaces due to the torque is quite intense and wear develops very quickly, whereupon the clutch becomes noisy. When the multiple disc clutch first came into use three or four studs were frequently used as driving means, because this is a relatively inexpensive construction, but after the tendency of this design to become noisy quickly had come to light, it became the practice to cut both the drums and the discs with (involute) gear teeth, which multiplied the area of the contact surface many times.

Lever-operated Clutch—In farm tractors the engine power is used both for propulsion and for operating machinery through a belt pulley, and in some designs the engine clutch is used for controlling the power for both purposes. It is then necessary that the clutch should be controllable from the ground as well as from the driver's seat, which calls for lever control.

A clutch designed for such work is the Twin Disc illustrated in Fig. 21. It has a single driving plate sliding on studs fastened into the flywheel, and two driven plates, of which one is keyed to a sleeve, against a collar on same,

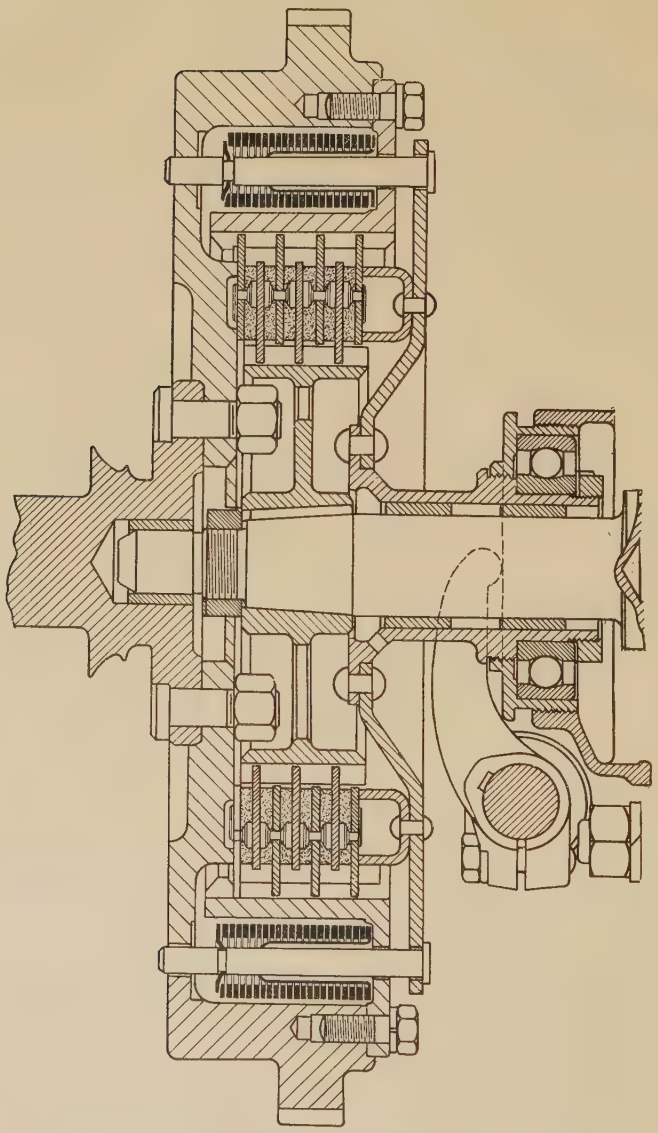


Fig. 20.—Multiple-Disc, Dry Clutch (Packard).

while the other slides on dowels in the hub of the former. A collar screwed over the sleeve has a number of levers fulcrumed on it, these levers being operated by a sliding cone on the sleeve. The contact surfaces of the levers are hardened, and they bear against the hardened cone and hardened pins set into the rearmost disc. Two floating discs of friction material are interposed between the metal discs.

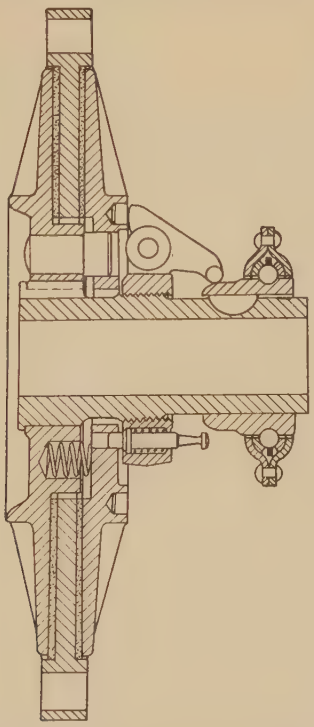


Fig. 21.—Hand-Operated Clutch for Tractors.

A clutch of this kind naturally requires very fine adjustment, and this is provided for by a lock pin in the threaded collar and twelve holes near the inner edge of the adjacent disc. The lock pin can be withdrawn against the pressure of a light spring, and by then turning the threaded collar through one twelfth of a revolution and allowing the lock pin to enter the next adjusting hole, an adjustment of 0.005 in. can be made. End thrust is taken up entirely within the clutch itself and is not transmitted to the crankshaft of the engine.

CHAPTER III

Some Principles of Toothed Gearing

In the transmission of power from the engine to the driving wheels of a vehicle, extensive use is made of toothed gearing. Automotive gearing must be unusually compact, and in addition it must be silent and efficient in operation. These are more or less conflicting requirements, and in order that the best possible results may be obtained it is important that the designer be familiar with the underlying principles of such gearing.

The Involute—The tooth form universally used in automotive work is that of the involute. An involute is a curve formed by the end of a cord or tape when the latter is unwound from a circle, as shown in Fig. 1. The circle from which the generating cord is unwound is known as the base circle. It will be seen that the involute curve originates at and lies wholly outside the base circle, and it meets the base circle at right angles.

Fig. 2 will serve to make clear one of the valuable properties of the involute curve. The two large circular arcs are portions of the base circles of two mating gears. Suppose a cord of no material thickness to be wound around these two circles in opposite directions, so it can be unwound from one circle and wound up on the other. This cord is represented in the drawing by the straight line X-Y, which is tangent to both circles. When the cord is thus wound from one circle onto the other, any point in it, such as A, moves along a straight line in space. But with relation to the base circle B_1 it describes a portion of an involute curve, which is shown in dashed line. Its motion relative to base circle B_2 is also a portion of an involute curve, which is shown dotted. Point A is common to both involutes and a point of tangency throughout its motion from X to Y.

If the cord is unwound from base circle B_2 onto base circle B_1, motion is transmitted from B_2 to B_1 at a uniform rate. That is, if B_2 turns at uniform angular speed, B_1 also will turn at uniform angular speed. If now the two circles

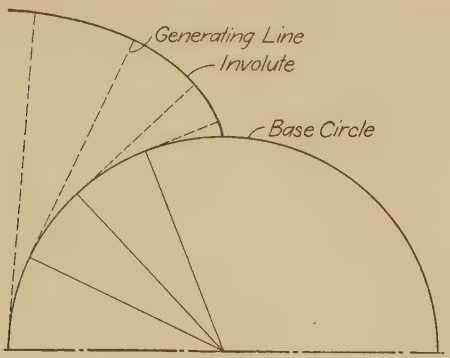


FIG. 1.—THE INVOLUTE CURVE AND ITS GENERATION.

(in the form of discs) are provided with involute "cams" of the shape shown by the dashed and dotted curves, these cams, by contacting with each other, will transmit motion from B_2 to B_1 in exactly the same way as it is transmitted by the cord X-Y. If the angular speed of B_2 is constant, that of B_1 also will be constant. Such transmission of motion at a constant ratio is necessary in gearing in order to prevent shock and noise. If the motion is transmitted by the involute "cams" (which are equivalent to gear teeth) the point of contact between them is at all times in the straight line X-Y, which is known as the line of action.

Pitch—In the calculation of gears and gear teeth the pitch circle plays an important part. This is an imaginary circle perpendicular to the axis of the gear, at the points of intersection of the circumference of which with the tooth faces there is no sliding action when the gear runs with

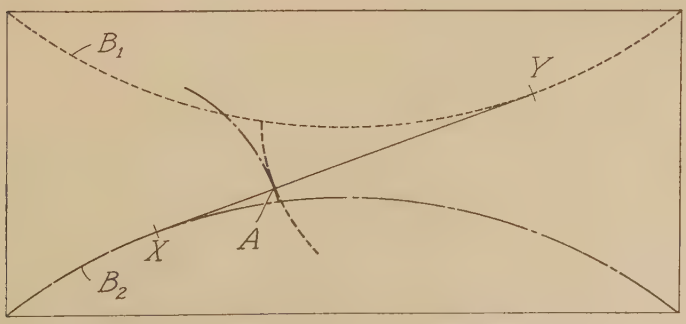


FIG. 2.—DIAGRAM SHOWING LINE OF ACTION.

another (both being assumed to be correctly dimensioned, aligned and spaced). A somewhat simpler definition can be given by reference to two mating gears. The pitch circles of two correctly meshed spur gears are equal, respectively, to the cross-section circles of two friction rollers on the same axes which—without slippage—give the same speed ratio as the pair of gears. This is illustrated in Fig. 3. The pitch circle of a gear is always larger than its base circle.

Pressure Angle—The pitch diameter of a gear is the diameter of its pitch circle. The pitch point is the point of tangency of the two pitch circles. The pressure angle (θ, Fig. 4) of a gear tooth is the angle made by a tangent to the tooth profile at the pitch circle with a radial line from the

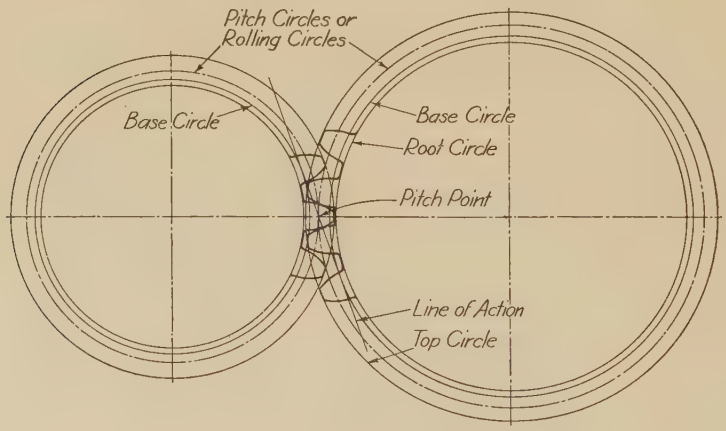


FIG. 3.—PITCH-CIRCLE DIAGRAM.

center of the pitch circle through this point (or between a line normal to the tooth profile at the pitch point and a common tangent to the two pitch circles). Previous to the advent of the automobile, a pressure angle of 14½° was practically standardized, and this angle is still used almost exclusively for so-called industrial gearing. For use in automotive gearing it has the disadvantage that in pinions with small numbers of teeth it leads to excessive "undercutting," which reduces the resistance of the teeth to bending stresses. Moreover, only the involute portion of the tooth profile, which in such a pinion extends only slightly below the pitch circle, is useful for tooth action, and the effective depth of tooth is therefore reduced.

As a result, for automotive purposes a 20° pressure angle was adopted. By referring to Fig. 1 it will be seen that, while the involute curve meets the base circle at right angles, if we pass along the curve from the base circle, the farther we proceed the greater becomes the angle at which the curve cuts a radial line from the center of the base circle. The point where the involute makes a 20° angle with a radial line is farther from the base circle than the point where it cuts a radial line at 14½°, which shows that with the former pressure angle the involute extends further below the pitch line. As may be seen from Fig. 4 the pitch radius and the

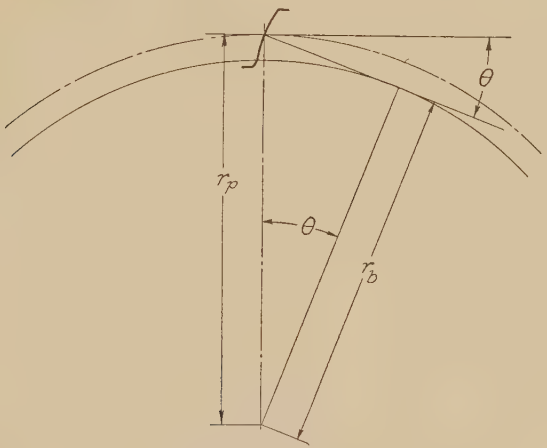


FIG. 4.—PRESSURE-ANGLE DIAGRAM.

radius of the base circle are connected by the following equation:

$$r_b = r_p \cos \theta.$$

Proportions of Gear Elements—The relative size of a gear tooth is expressed by its diametral pitch. Mathematically, the diametral pitch of a gear is the number of teeth per inch of pitch diameter. In the form of an equation this may be expressed as follows:

$$\text{Diametral pitch } (p_d) = \frac{\text{Number of teeth}}{\text{Pitch diameter}} = \frac{n}{d_p}$$

which may be changed to the form

$$\text{Pitch diameter } (d_p) = \frac{\text{Number of teeth}}{\text{Diametral pitch}} = \frac{n}{p_d}.$$

The circular pitch of a gear is the distance from the face of one tooth to the corresponding face of an adjacent tooth, measured along the pitch circle. Since the length of the pitch circle is πd_p, the circular pitch evidently is

$$p_c = \pi d_p / n;$$

and since, as shown above,

$$d_p = n/p_d,$$

substituting this value for d_p in the above we get

$$p_c = \pi / p_d.$$

The radial distance from the pitch circle to the top of the teeth (or the top circle) is known as the addendum, and the radial distance from the pitch circle to the root circle is known as the dedendum. The working depth is equal to twice the addendum, and the dedendum is equal to the sum of the working depth of the tooth below the pitch circle and of the clearance. For so-called "full depth" teeth the following relations hold:

Addendum	= 1/Diametral pitch;
Dedendum	= 1.157/Diametral pitch;
Working depth	= 2/Diametral pitch;
Clearance	= 0.157/Diametral pitch;
Outside diameter of gear	= $(n + 2)$/Diametral pitch.

Stub Teeth—For use in automotive and similar work, a shortened form of tooth has been widely used, known as the stub tooth. It was first introduced by the Fellows Gear Shaper Co. and generally has a 20° pressure angle. Such gears are designated by double pitch numbers as 6–8, which means that the circumferential dimensions of the teeth, such as the circular pitch and thickness of tooth, correspond to a 6 diametral pitch, whereas the radial dimensions, that is, the addendum, dedendum and clearance, correspond to an 8 diametral pitch. Pitches in common use in the automotive industry are 5–7, 6–8, 7–9 and 8–10. One disadvantage of this system is that the proportions between the radial and circumferential elements of the teeth vary with the pitch.

The American Gear Manufacturers Association has recently adopted a standard for a 20° stub involute system

which is now slated for adoption as an American standard and is likely to come into use in automotive practice in the near future. In this system the tooth elements have the following proportions:

Addendum	= 0.8/Diametral pitch;
Minimum dedendum	= 1/Diametral pitch;
Working depth	= 1.6/Diametral pitch;
Minimum total depth	= 1.8/Diametral pitch.

Strength of Gears—Gears may fail in at least three different ways, and must be designed so they are capable of withstanding the three kinds of destructive effects. In clash gears, the edges of the tooth faces at which engagement takes place may be battered up by the clashing. This, however, is best guarded against by reducing the shock of engagement, by lessening the spinning weight. Secondly, the teeth may fail by breakage, and to prevent this they must be made of adequate strength. This leads to one method of gear calculation, that based on the resistance of the teeth to breakage under bending stresses.

The third kind of failure is by wear of the tooth faces. Such wear takes place principally at and near the pitch line; it destroys the correct tooth form and causes the gears to operate noisily. Gear materials and methods of heat treatment have been constantly improved, and cases of tooth breakage are very rare. On the other hand, the aim in the design of transmissions has been constantly to make them more compact, especially those intended for passenger cars; and, as a result, the unit pressures on the gear teeth are intense, which often leads to pitting of the teeth, followed by rapid wear. Calculation of the gears on the basis of their resistance to wear is therefore really more important than calculation on the basis of strength.

Lewis Formula—The torque which can be transmitted by a gear without danger of breakage of its teeth is usually calculated by means of a formula first given by Wilfred Lewis in a paper presented by him before the Engineers' Club of Philadelphia in 1893. The formula reads,

$$W = S\,p\,f\,y,$$

where W is the tangential force in pounds on the pitch circle of the gear; S, the stress on the material of the teeth, in lbs. per sq. in.; p, the circular pitch in in.; f, the face of the gear in in.; and y, a constant depending upon the form of the teeth, which varies with the number of teeth in the gear and the proportion of the addendum. The stress due

to bending moments is always greatest in the teeth of that member of a pair which has the smallest number of teeth—the pinion—and it is this to which the strength calculations are applied.

Lewis assumed that the whole of the tangential force acts on a single tooth, and at its extreme end. The actual pressure on the tooth is not in the direction of a tangent to the top circle but makes an angle equal to the pressure angle with that tangent—if we neglect the friction component, which is very small in well-lubricated gears. The tooth is then considered as a cantilever, and its strength is determined by the base width and the height of a parabola which can be inscribed in the tooth section, with its apex at the point where the line of force intersects the radial center line of the tooth. The parabola represents a cantilever which, when a trans-

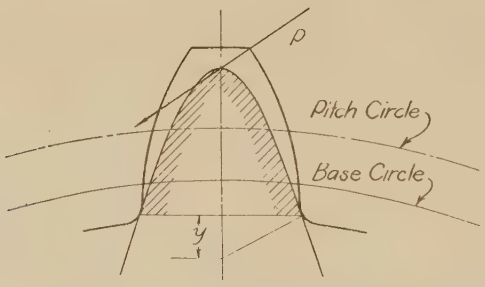


FIG. 5.—UNIFORM-STRESS PARABOLA INSCRIBED IN GEAR-TOOTH OUTLINE.

verse force is applied at its free end, is equally stressed at all sections, and that section of the tooth where the parabola is tangent to its profile is therefore the weakest. (See Fig. 5.)

The assumption that the whole of the tangential force comes on a single tooth and at the very end thereof is not warranted in the case of accurately made gearing, although it may have been justified in the case of the less accurate gears of the past century. The result is that when the above equation is used, the safe stress S can be given unusually high values. The safe stress, moreover, varies with the pitch line velocity, as when gears are in operation, dynamic loads due to inaccuracies of tooth spacing and contour are added to the static loads, which alone are considered in the original Lewis formula. These dynamic loads are usually taken care of by means of a load increment factor of the form $(v + c)/c$, where v is the pitch line velocity and c a constant.

Values of the Lewis constant y for different forms and numbers of teeth are given in the following table:

Table I—Values of the Tooth-form Factor y

No. of Teeth	15° Full Depth	20° Full Depth	20° T. A.* Standard	20° 5–7 Pitch	20° 7–9 Pitch
14	0.072	0.088	0.098	0.110	0.101
15	0.075	0.092	0.102	0.114	0.106
16	0.077	0.094	0.105	0.118	0.110
17	0.080	0.096	0.110	0.122	0.114
18	0.083	0.098	0.113	0.125	0.118
19	0.087	0.100	0.116	0.127	0.121
20	0.090	0.102	0.119	0.130	0.123
21	0.092	0.104	0.121	0.132	0.125
23	0.094	0.106	0.124	0.136	0.129
25	0.097	0.108	0.129	0.139	0.132
27	0.100	0.111	0.131	0.142	0.136
30	0.102	0.114	0.135	0.144	0.138
33	0.103	0.117	0.137	0.147	0.140
36	0.105	0.120	0.139	0.149	0.143
39	0.107	0.123	0.141	0.150	0.144
42	0.109	0.125	0.142	0.152	0.146
46	0.111	0.128	0.144	0.153	0.148
50	0.112	0.130	0.146	0.155	0.149
Rack	0.146	0.154	0.169	0.177	0.172

*This column also applies to 4–5 pitch and 8–10 pitch 20° involute gears.

Compressive Stresses on Gear Teeth—Prof. H. Hertz, who developed the mathematical theory of the compressibility of elastic bodies, found that the maximum compressive stress produced when two cylinders are pressed together on their cylindrical surfaces by a force P_1, is given by the equation

$$S_c = \frac{4P_1}{\pi bl},$$

where b is the width of the area of contact and l its length. For the width b of the area of contact he found the following expression:

$$b = 4\sqrt{\frac{2P_1}{\pi lE(1 - 1/m^2)(1/r_1 + 1/r_2)}}$$

where E is the coefficient of elasticity of the material of the cylinders; m, the ratio of longitudinal contraction to lateral expansion under compressive loads, and r_1 and r_2 are the radii of the two cylinders. The factor m usually has a value

of about 3.5, so that $1-1/m^2$ is about 0.92. Inserting this value in the expression for b we get

$$b = 4\sqrt{\frac{2.174P_1}{\pi lE(1/r_1 + 1/r_2)}}.$$

By now substituting this value of b in the expression for S_c we get

$$S_c = \frac{P_1}{\pi l}\sqrt{\frac{\pi lE(1/r_1 + 1/r_2)}{2.174P_1}}$$

which after simplification reduces to

$$\sqrt{\frac{P_1E(1/r_1 + 1/r_2)}{7l}}.$$

In the case of a pair of gear teeth in contact at the pitch point—which is the most important point, since the pressure concentration is highest there—the pressure P_1 forcing the teeth together can be expressed in terms of the tangential force P on the pitch circle and of the pressure angle; and the radii of curvature of the contact surfaces, r_1 and r_2, can be expressed in terms of the pitch radii r and R of the pinion and gear, respectively, and of the pressure angle. As may be seen from Fig. 6,

$$P_1 = P/\cos\theta. \quad r_1 = r\sin\theta. \quad r_2 = R\sin\theta.$$

From this we find that

$$(1/r_1 + 1/r_2) = (r + R)/rR\sin\theta.$$

Now let $R = \rho r$,

then

$$\frac{r + R}{rR\sin\theta} = \frac{\rho + 1}{R\sin\theta}.$$

Substituting this value in the above equation for S_c as well as $P/\cos\theta$ for P_1 and f, the width of the gear face, for l, the length of the cylinder, we get

$$S_c = \sqrt{\frac{PE(\rho + 1)}{7fR\sin\theta\cos\theta}}.$$

But since $\sin\theta\cos\theta = \frac{1}{2}\sin 2\theta$, this can be simplified to

$$S_c = \sqrt{\frac{PE(\rho + 1)}{3.5fR\sin 2\theta}}$$

where P is the load in lbs. per sq. in. on the pitch line; E, the coefficient of elasticity of the material (30,000,000 for steel); ρ, the ratio of the number of teeth in the gear to the number of teeth in the pinion; f, the face width of the gears in inches; R, the pitch radius of the gear (the larger of the two mating gears), in inches, and θ, the pressure angle.

Since in automotive work gears with a pressure angle of

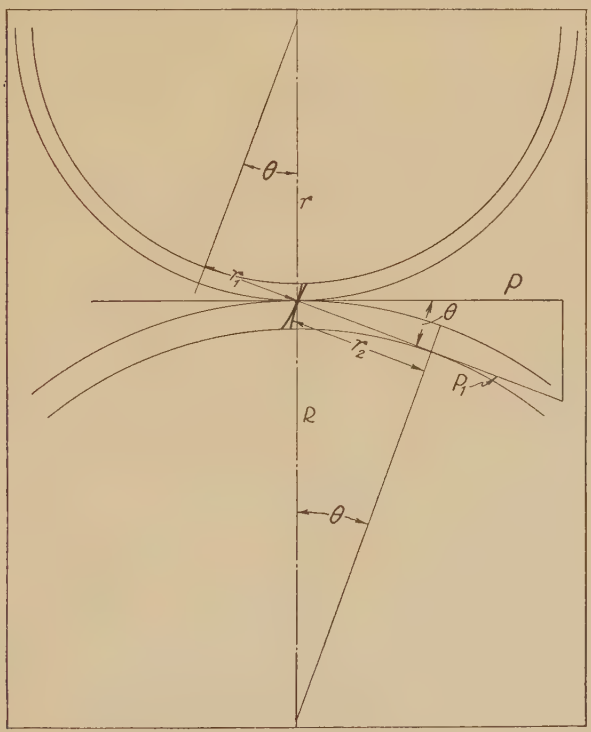


FIG. 6.—DIAGRAM FOR RADII OF TOOTH-CONTACT SURFACES.

20° are used almost exclusively, by inserting the value of sin 40° and taking all of the known values out from under the radical sign, we get for the maximum compression stress on the teeth of 20° pressure angle gears

$$S_c = 3900\sqrt{\frac{P(1+\rho)}{fR}} \text{ lbs. per sq. in.}$$

Compressive Strengths of Gear Materials—Not many data as to the compression strengths of different materials are

available, but Prof. Stribeck, who did some of the pioneer work in the development of annular ball bearings, found that the chrome steel used for the balls of such bearings, with one particular heat treatment had an ultimate strength under compression of 648,000 lbs. per sq. in.; and with another, 685,000 lbs. If the above equation gave the actual limiting compression stress on the teeth, the way gears are usually proportioned, they would have a very considerable factor of safety. However, the stress calculated is the stress under static loads, and the dynamic loads obtaining when the gears are being operated at high speeds are materially higher. It is customary in gear practice to allow for the effect of dynamic loads by using a load increment factor. However, in any particular class of automotive work the pitchline velocities do not vary greatly, and this load increment may be allowed for by suitably increasing the factor of safety or by lowering the allowable compressive stress S figured with.

Number of Teeth in Contact—In order that tooth action may be smooth and continuous, each pair of teeth must come into contact before the preceding pair has broken contact. In fact, the greater the overlap of periods of tooth contact the smoother the action.

As already explained, contact takes place along the line of action, a straight line through the pitch point and tangent to the two base circles. As shown in Fig 7, tooth contact begins at the point of intersection B of the line of action with the top circle of the driven gear, and ends at the point of intersection C of the line of action with the top circle of the driving gear.

With tooth numbers within the range usually employed in automobile transmissions, there is at all times at least one pair of teeth in contact, and during a fraction of the time a second pair is in contact. This is expressed by saying that the average number of pairs of teeth in contact is a compound number having a value of between one and two. As will be shown later on, while the pinion turns through an arc corresponding to one circular pitch, the contact point travels along the line of action a distance equal to one normal pitch (or base-circle pitch), and the number of pairs of teeth in contact is found by dividing the length of the line of contact by the normal pitch. To calculate the number of pairs of teeth in contact we must therefore first determine the length of the line of action, to which end we proceed as follows:

Referring to Fig. 7, the line of contact is represented by BC, B being the point where contact begins and C the point

where it ends. P is the pitch point. The addendum is equal to c/p_d, where c is a constant varying with the type of gearing ($c = 1$ for full depth teeth; 0.8 for Tentative American standard stub teeth and 4–5 and 8–10 pitch Fellows stub teeth;

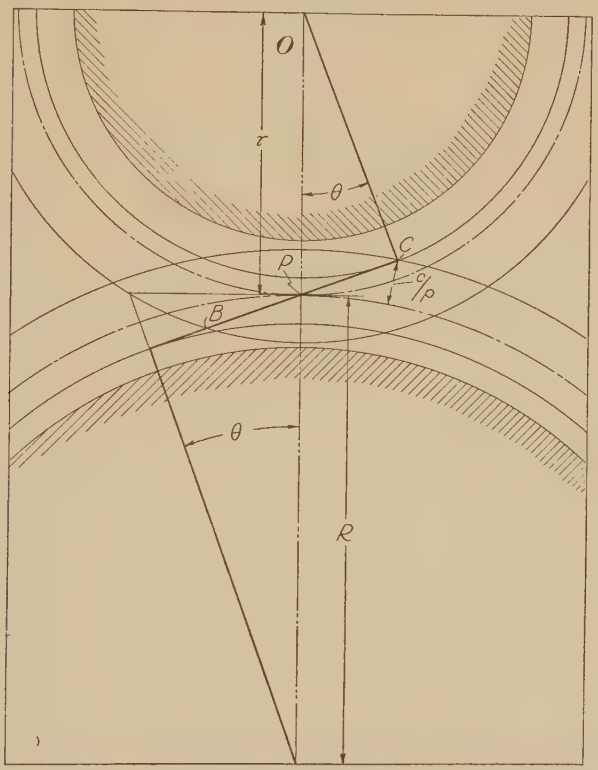


FIG. 7.—DIAGRAM FOR THE DETERMINATION OF THE NUMBER OF TEETH IN CONTACT.

(The lower, larger gear is the driver and turns right-handedly.)

0.715 for 5–7 pitch and 0.78 for 7–9 pitch Fellows stub teeth).

$$BP = BC - PC \quad \text{and} \quad PC = r \sin \theta,$$

where r is the pitch radius of the driven member

$$BC = \sqrt{OB^2 - OC^2}$$

$$OB = r + (c/p_d)$$

$$OC = r \cos \theta$$

Hence

$$BC = \sqrt{(r + c/p_d)^2 - (r \cos \theta)^2}$$

and

$$BP = \sqrt{(r + c/p_d)^2 - (r \cos \theta)^2} - r \sin \theta.$$

Now,

$$r = n/(2p_d)$$

and substituting this in the above equation we get

$$BP = \sqrt{\left(\frac{n + 2c}{2p_d}\right)^2 - \left(\frac{n \cos \theta}{2p_d}\right)^2} - \frac{n \sin \theta}{2p_d}.$$

$$= \tfrac{1}{2}p_d\sqrt{(n + 2c)^2 - (n \cos \theta)^2} - n \sin \theta.$$

In an exactly similar way it is found that

$$PC = \tfrac{1}{2}p_d\sqrt{(N + 2c)^2 - (N \cos \theta)^2} - N \sin \theta.$$

The total length of contact is

$$BC = BP + PC$$

$$= \tfrac{1}{2}p_d[\sqrt{(N + 2c)^2 - (N \cos \theta)^2}$$

$$+ \sqrt{(n + 2c)^2 - (n \cos \theta)^2} - (N + n) \sin\theta].$$

The number of teeth in simultaneous contact may now be determined from the following consideration. Referring to Fig. 8, a tooth of the pinion or driving gear is about to break contact at B, and the line of action may therefore be considered as part of the generating line of the involute constituting the active face of this particular tooth. Therefore, if we wind the generating line back onto the base circle, the point D where B meets the base circle may be taken to denote the angular position of the pinion at which contact is broken, whereas the point A denotes the angular position where contact begins. If the distance between these two points along the base circle were exactly equal to the normal pitch, that is, to the distance along the base circle from the face of one tooth to the corresponding face of the next, then, evidently, as one pair of teeth broke contact, the next one would come into contact, and one pair of teeth and no more would be in contact at all times. Actually the distance between these two points is always greater than one normal pitch, and the number of pairs of teeth in simultaneous contact is

found by dividing the length of the line of action by the normal pitch, which latter is equal to $\pi \cos \theta / p_d$. But dividing by $\pi \cos \theta / p_d$ is equal to multiplying by $p_d(\pi \cos \theta)$ hence, multiplying the expression for the length of the line of action by the expression for the normal pitch, we get for the number of pairs of teeth in simultaneous contact:

$$\frac{1}{2\pi \cos \theta}[\sqrt{(N + 2c)^2 - (N \cos \theta)^2} + \sqrt{(n + 2c)^2 - (n \cos \theta)^2} - (N + n) \sin \theta].$$

It will be seen from this equation that the number of pairs of teeth in contact is independent of the pitch and depends only on the numbers of teeth, the pressure angle, and the proportion of the working depth.

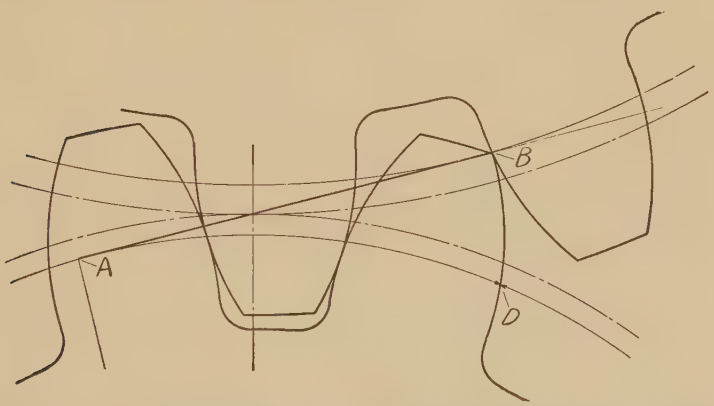


FIG. 8.—SHOWING POINTS WHERE TOOTH-CONTACT BEGINS AND ENDS.

Tooth Modification—In practice the number of pairs of teeth in contact is often reduced by a modification of the tooth profile. While accurately cut involute gears would run together smoothly if they were mounted on correct centers and the material of both the gears and the supporting members were absolutely rigid, in practice these conditions are not fulfilled, and there is a tendency for the outer end of the gear teeth to dig into the teeth of the mating pinion, causing noise and roughness of action. To prevent this, the contour of the teeth is suitably modified. The modified portion of the tooth face, which is near the top of the tooth and may extend over one quarter the addendum, is usually also

an involute, but of greater pressure angle than the working portion of the tooth face.

Sample Calculations—To give an idea of the pairs of teeth in contact in gears of the proportions used in automotive transmissions, we will apply the equation developed above to a pair of gears with 17 and 29 teeth, of 7–9 pitch. We then find the number of pairs of teeth in contact to be 1.34. If instead of 7–9 pitch teeth we had chosen 20° pressure-angle, full-depth teeth, the number of pairs in contact would have worked out to 1.57, and if we had chosen, instead of 17 and 29 7–9 pitch, 20 and 33 8–10 pitch teeth, which require about the same center distance, the number of pairs in contact would have figured out to 1.62. This latter gain may seem surprisingly low in view of the considerable increase in the number of teeth. A great improvement in the smooth-

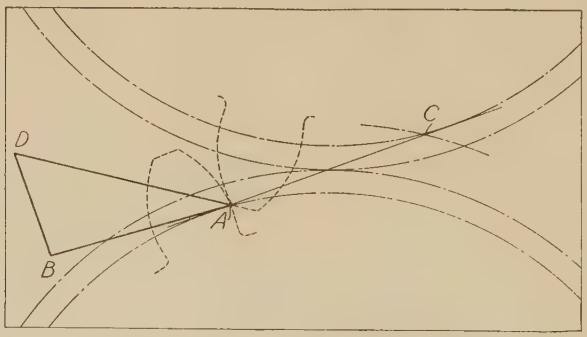


FIG. 9.—RUBBING-SPEED DIAGRAM.

ness of action is obtained if the number of pairs of teeth in contact is increased to above 2. Then the tooth load is at all times divided between two teeth (not equally, it is true), whereas if the number of teeth in contact is less than 2, the full tooth load is taken by a single tooth for an instant.

Rubbing Speed of Teeth—The rubbing speed or rate of sliding motion between gear teeth at any point of the line of action may be readily found graphically. Referring to Fig. 9, let us take the point *A* at which contact begins, the point of intersection of the gear top circle with the common tangent to the two base circles. From the speed of revolution of the pinion and its base circle diameter we can readily find the velocity of this point, which we may represent by a line *BA* tangential to the base circle at point *A*. We next measure the distance of the point of contact on the tooth of the driven

gear, from the axis of that gear, which gives us the radius of the circle described by this point; and with this and the speed of revolution of the driven gear we can calculate the velocity of this point, which we represent by a straight line *DA* tangent to the circle described by the point in question. Both velocities, of course, are plotted to the same scale. Then the line *DB*, connecting the two lines already drawn, represents the sliding motion between teeth, in magnitude as well as in direction.

By dividing the line of action into a number of equal parts, we can determine the speed of sliding motion between teeth at each division point, and we can then plot the values obtained, plotting distances along the line of action on the

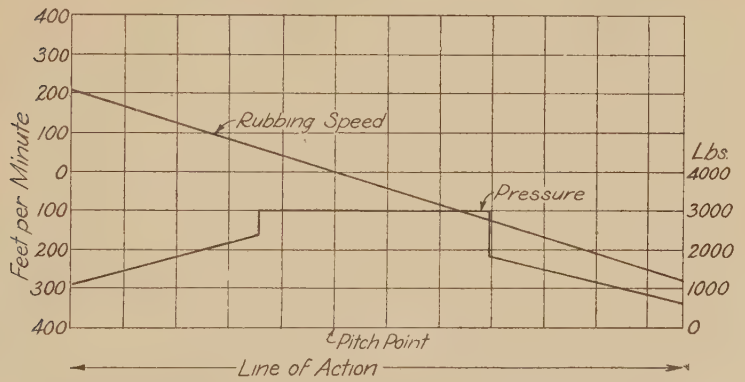


FIG. 10.—GRAPH OF RUBBING SPEED AND TOOTH PRESSURE.

axis of abscissas, and the corresponding rubbing speeds on the axis of ordinates. When the teeth are in contact at the pitch point, there is no sliding motion, as the tangents to the circles described by the two contact points then coincide. This gives a graph of the form of Fig. 10 (which is taken from an article on Spur Gears by J. W. Snarry in "The Automobile Engineer," for December, 1922). It will be seen that the rubbing speed varies directly with the distance along the line of action. It becomes zero at the pitch point, and the sliding motion then reverses in direction. The values given in Fig. 10 correspond to a pair of 7-pitch $14\frac{1}{2}°$ pressure-angle, 15 and 30-tooth gears with the pinion running at 1000 r.p.m. The axis at the left corresponds to the base line of the pinion, and it will be seen that the rubbing speed is less there than near the top of the pinion tooth.

Distribution of Tooth Load—It has been pointed out that Lewis, in deriving his formula for the strength of gear teeth, assumed that the whole of the tooth load came on a single tooth and on the extreme outer end of same. This assumption errs on the side of safety. To assure smooth action, the average number of pairs of teeth in contact must be greater than one, as previously pointed out, and if this is so, when one particular tooth makes contact either near its outer edge or its base line, a second tooth of the same gear also is in contact with its mate. The whole tooth load is taken by a single tooth only when the contact is near the pitch line. If the gears have been laid out to scale, that part of the line of action over which the entire tooth load is taken by a single tooth can be easily determined by laying off from each end of the line of action a distance equal to half a normal pitch. This has been done in Fig. 10, where two pairs of teeth are in contact over those portions of the line of action for which the pressure line is inclined.

As long as two teeth of the pinion are in contact the load is shared by them in the proportion of the effective stiffnesses of the respective pairs of teeth, which differ only because the loads are applied at different distances from the root. The teeth forming cantilevers, for the same deflection they will support loads proportional, to $L_1^3 + L_2^3$ (Fig. 11). Just before one of the teeth passes out of contact it still carries considerable load, and this load is then suddenly transferred to the other tooth. The variation of the load on one tooth throughout its period of contact is somewhat as shown in Fig. 10. The sudden increase in the load on one tooth when the other breaks contact is the cause of pitting. The load naturally is most intense on the pinion tooth, because of the smaller radius of curvature of that tooth, and when the sudden increase in load on the pinion tooth occurs, the contact is slightly below the pitchline, which is precisely the point where pitting generally shows up first.

Long and Short Addendum Gears.—Toothed gearing has long been standardized to a considerable extent, not through any action of some recognized standardizing body, but through the influence of a number of firms in the gear-cutting machinery industry. One pressure angle is used for a very large proportion of all industrial gearing, there is an accepted series of diametral pitches, and fixed proportions are practically always used for the addendum and dedendum. This standardization has been highly beneficial in connection with work turned out in small quantities, as it obviates making special cutters for each job. But in

the case of automobile transmissions made in large quantities there is no particular advantage from the production standpoint in adhering to these standards, and all of the factors mentioned may be varied as desired if any gain in the operation of the gears is obtainable thereby.

It has been mentioned already that the active face of a gear tooth extends down only to the base circle, which in gears with small numbers of teeth is much closer to the pitch circle than the length of the addendum. That portion of the tooth flank which lies below the base circle and extends down to the clearance circle is ordinarily a straight radial line. However, with a pinion having a small number of teeth, in order to prevent interference between its teeth and those of

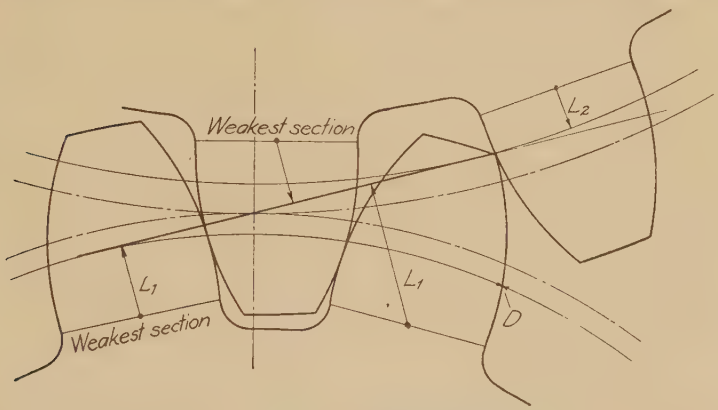


FIG. 11.—SHOWING VARIATION OF EFFECTIVE LEVER ARM OF TEETH DURING PERIOD OF ACTION.

the gear meshing with it, it is necessary to cut below this radial line, and this is known as undercut. What is meant by interference is made clear by Fig. 12, which shows a portion of a 12-tooth pinion with 14½° full-depth involute teeth in mesh with rack teeth. The black-sectioned areas represent interference. It is apparent from the drawing that interference results from the fact that the radial part of the pinion-tooth flank makes a larger angle with the radial line through the pitch point (line of centers) than does the side of the rack tooth. The angle which the side of the rack tooth makes with the line of centers is constant (being equal to the pressure angle), while the angle made by the radial part of the pinion-tooth flank with the line of centers increases as the end of the line of action is approached. Therefore,

with any given pinion, if there is interference at all, there will be interference just previous to the breaking of tooth contact. Moreover, the angle made by the radial part of the tooth flank with the line of centers at the moment tooth contact is broken decreases as the number of teeth increases.

Minimum Number of Teeth Without Interference—With increase in the number of pinion teeth interference ceases when, with the pinion in the position where tooth contact theoretically ceases, the side of the rack tooth in contact with the pinion tooth, and the radial portion of the tooth flank both make the same angle with the line of centers;

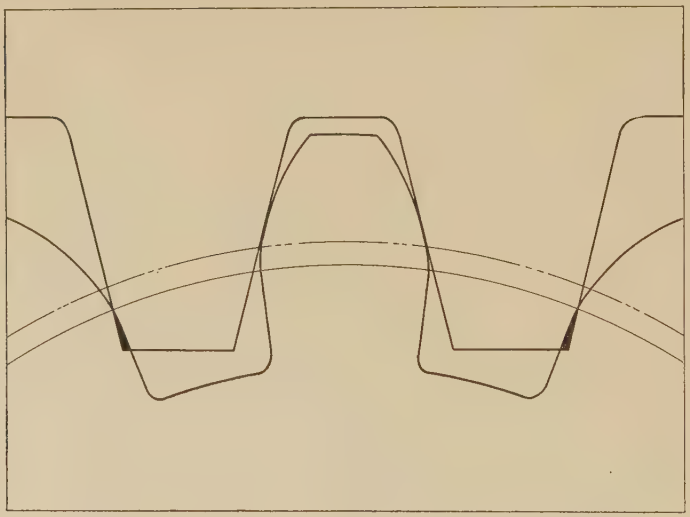


FIG. 12.—SHOWING INTERFERENCE BETWEEN TEETH

and since there is contact between these two lines, this involves that they coincide in part. This enables us to calculate the minimum numbers of teeth for involute gears with different pressure angles and different proportional addenda for which there is no interference, and hence no undercutting.

In Fig. 13 is shown a pinion with 20° full-depth involute teeth, in mesh with a rack, and contact between two teeth has just come to an end. The side of the rack tooth and the radial portion of the pinion-tooth flank are shown to coincide partly, hence there can be no interference. From the geometry of the drawing we obtain the following:

$$AB = R \sin \theta. \qquad BC = AB \sin \theta.$$

Hence $$BC = R \sin^2 \theta.$$

But BC is the addendum of the rack tooth, which (for a full-depth tooth) is equal to $1/p_d$.

By definition we have

$$p_d = n/d_p = n/2R$$

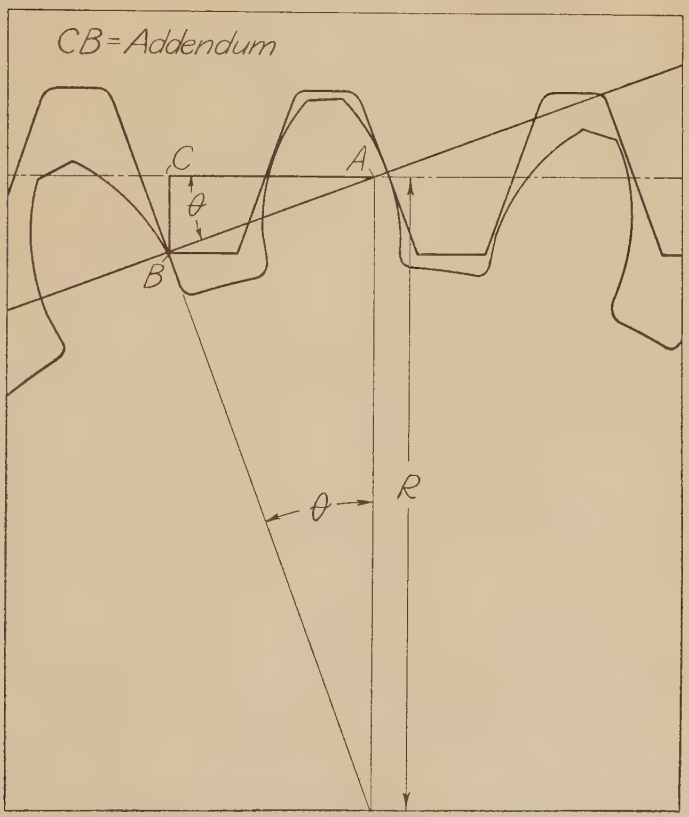


FIG. 13.—GEOMETRICAL RELATIONS DETERMINING MINIMUM NUMBER OF TEETH WITH WHICH THERE IS NO UNDERCUTTING

where n is the number of teeth in the pinion and R the pitch radius. Hence,

$$1/p_d = 2R/n.$$

Therefore

$$2R/n = R \sin^2 \theta$$

and

$$n = 2/\sin^2 \theta.$$

This applies to full-depth teeth. In the case of stub gears with teeth whose depth bears a ratio r to the full depth, the minimum number of teeth for which there is no undercutting is

$$n = 2\,r/\sin^2 \theta$$

With the aid of these equations we find that the minimum number of teeth which completely eliminates undercutting is 33 for a 14½° full-depth pinion, and 18 for a 20° full-depth pinion, although with 32 and 17 teeth, respectively, the undercut is negligible. For a 20° stub-tooth pinion with an addendum of 0.8 the normal, the number is 14.

With formed cutters of the milling cutter type it is impossible to produce an undercut, hence the gears cut by this process are not plain involute gears; only the central portion of the tooth profile is an involute, while the ends are cycloids (curves generated by one circle rolling on another).

Diameter Increments and Decrements to Obviate Undercut—It has been suggested that in pairs of gears comprising a pinion with a relatively small number of teeth, the addendum of the pinion should be increased and the addendum of the gear decreased an equal amount to assure complete involute action. To increase the pinion addendum, the outside diameter of the pinion blank must be made larger than according to the usual formula, and the outside diameter of the gear blank must be made equally smaller. We can readily develop the above equation for the relation between number of teeth, pressure angle and addendum that assures full involute action, so as to give directly the increments and decrements on outside diameters or blank diameters required with any number of pinion teeth and any pressure angle to assure full involute action.

Since

$$n = 2r/\sin^2 \theta, \quad r = n \sin^2 \theta/2.$$

If instead of making the gear addendum $1/p_d$ we make it r/p_d (where r is less than unity), it is reduced by $(1 - r)/p_d$, and as the outside diameter of the blank will be reduced by twice this amount, the diameter decrement will be $2(1 - r)/p_d$. Inserting the value of r in terms of the number of pinion teeth and pressure angle (as found above) we get

$$D = (2 - n \sin^2 \theta)/p_d.$$

For instance, in order that there may be complete involute

action in a pair of full-depth 6-pitch 20° involute gears comprising a 14-tooth pinion, the diameter of the pinion blank must be increased—and that of the gear blank decreased—by

$$(2 - 14 \times \sin^2 20°)/6 = 0.0604 \text{ in.}$$

This assures complete involute action so far as the pinion is concerned, but if the gear had only a few more teeth than the pinion, its teeth might have to be undercut in order to avoid interference with the long-addendum pinion-teeth. That this may not be necessary, the sum of the numbers of teeth in pinion and gear must be at least 34 in the case of 20° full-depth teeth and 64 in the case of 14½° full-depth teeth.

Long and short addendum teeth have been extensively used in bevel gears for automobile final drives. In that case it has been the practice to make the addendum of the pinion a fixed fraction of the working depth, generally 0.7, the addendum of the gear teeth being made 0.3 this depth.

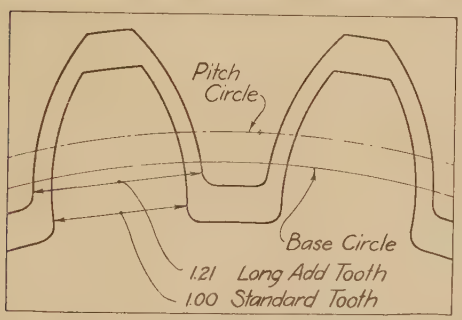


FIG. 14.—STANDARD AND LONG-ADDENDUM FULL-DEPTH TEETH COMPARED.

Figure 14 furnishes a comparison of the teeth of a standard 20-tooth 14½° pressure-angle, full-depth pinion with a corresponding pinion with long-addendum teeth, the addendum in this case being equal to 75 per cent of the working depth.

Smooth Action of Long-and-Short-Addendum Gears—It is claimed for long-and-short-addendum gears that they run together under load much more smoothly than gears in which the addenda are equal. The reason for this is that as the addendum of the pinion is increased, the arc of recess is increased, and as the addendum of the driven gear is decreased, the arc of approach is decreased. Consequently, we may say that in gears of which the pinion or driving member has a relatively long addendum, the arc of approach is shortened and the arc of recess lengthened. Now, it is a well-known fact that tooth action is not nearly as smooth during the approach as during the recess. There is sliding motion during each of these periods, but the two motions are opposite in direction and the actions during the two periods

have been likened, respectively, to pushing a long, flexible rod ahead of one over a rough road, and drawing the same rod behind one over the same surface, which latter action naturally proceeds much more smoothly.

Determining Circular Thickness for Long-and-Short-Addendum Gears—When the addendum and dedendum are unequal, the circular thickness of the tooth is no longer equal to half the circular pitch. How a variation of the proportion between the addendum and dedendum affects the circular thickness of the tooth can be readily seen from Fig. 15, which shows a rack tooth. At the normal pitch line the circular thickness is equal to half the circular pitch, and at the chosen pitch line the circular thickness is equal to one-half the circular pitch plus twice the length BC. Now

$$BC = AC \tan \theta$$

where θ is the pressure angle, and the length AC is equal to the difference between the chosen addendum and the normal addendum. Therefore, the actual circular thickness of the pinion is equal to

$$\text{Normal circular thickness} + 2\,AC \tan \theta.$$

For actual measurements, of course, it is necessary to have the chordal thickness. The angle ϕ (Fig. 16) subtended at the axis of the pinion by the circular thickness is evidently

$$\phi = 360 t_c / 3.1416 d \text{ degrees}$$

where t_c is the circular thickness and d the pitch diameter. The chordal thickness is equal to

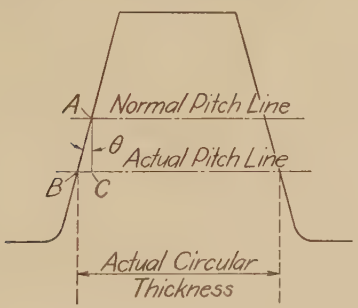
$$t_{ch} = d \times \sin (\phi/2).$$

The circular thickness of the gear tooth is equal to the circular pitch minus the circular thickness of the pinion tooth.

Helical Gears—In most cases where it is desirable that high speed gearing should operate silently, the gears are cut with teeth whose elements form helices instead of straight lines. One of the reasons for gear noises is deflection of the teeth under bending moments. Even without such deflection there is a sudden change in the tooth load when a pair of teeth ceases contact, and with deflection the sudden change in loading is even greater. With helical teeth a new pair of teeth does not come in contact suddenly over its whole width; contact begins at one end and gradually extends over the whole width, and the helix angle and the face width are always so proportioned that the leading edge of the tooth

comes in contact at the pitch line before the trailing edge of the preceding tooth has passed out of contact at the pitch line, hence the action is more nearly continuous, and causes of noise are minimized. Tooth action would be just as good with straight involute teeth if both the spacing and the contour of the teeth were perfect and the teeth were not deformed under load, but these conditions do not obtain in practice.

Helical gears are rarely used in sliding gear transmissions, for the reason that they produce considerable end thrust, and it is difficult to adequately provide for taking up this end thrust on suitable bearings. A number of transmissions are being built abroad with such gears, however, and this same type of gear is used also for engine front end drives. Herringbone gears, which are virtually two helical gears placed end



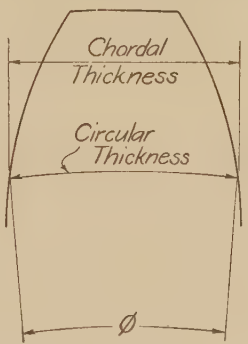


FIG. 15 (LEFT).—CIRCULAR-THICKNESS DIAGRAM.
FIG. 16 (RIGHT).—CHORDAL-THICKNESS DIAGRAM.

to end on the same axis, both gears being exactly alike except for the fact that the helices of the two are of opposite hand, have come into use recently in rear axles of the so-called double-reduction type.

In helical gears a distinction is made between the normal pitch and the apparent pitch. The apparent pitch is found by dividing the number of teeth by the pitch diameter, and the normal pitch is obtained by dividing the apparent pitch by the cosine of the helix angle. The helix angle is the angle made by a tangent to an element of the tooth in the pitch surface with an intersecting line parallel with the axis of the gear. The helical advance is the distance in the direction of the axis of the gear in which the elements of the tooth surface describe a complete circle. The following rules apply in the calculation of helical gears:

$$\text{Cosine of helix angle} = \frac{\text{Number of teeth}}{\text{Normal pitch} \times \text{pitch diameter}}.$$

$$\text{Helical advance} = \frac{3.1416 \times \text{pitch diameter}}{\text{Tangent of helix angle}}.$$

$$\text{Outside diameter} = \text{Pitch diameter} + \frac{c}{\text{Normal pitch}}.$$

where c is equal to 2 for full-depth teeth and 1.6 for the new standard stub teeth.

The addendum and dedendum are the same as in a spur gear of the same pitch as the normal pitch of the helical gear. Helical gears may be cut so as to properly mesh together when mounted on shafts making any desired angle with each other. The most common cases are where the shafts are parallel and where they are at right angles with each other, respectively. When the shafts are parallel, the helix angles must be the same for both pinion and gear, but one gear must be left-handed and the other right-handed. When the two shafts are at right angles with each other the sum of the helix angles of the two gears must be 90°, and both helices must be either right-hand or left-hand.

With helical gears mounted on parallel shafts, the helix angle is generally held down to 30°, in order to keep down the end thrust. The direction of the end thrust changes with the direction of the helices. When the shafts are at right angles, the helix angles of both gears generally are not far from 45°. The ratio between the speeds of revolution of two helical gears in mesh with each other is the reciprocal of the ratio of their respective numbers of teeth.

Metric Gear Standards—In countries employing the metric system of weights and measures, gear calculations are based upon the module, which corresponds to the reciprocal of our diametral pitch. All dimensions are in millimeters and the rules for spur gears are as follows:

$$\text{Module} = \frac{\text{Pitch diameter}}{\text{Number of teeth}}.$$

$$\text{Circumferential module} = 3.1416 \times \text{module}.$$

$$\text{Addendum} = \text{Module}$$

$$\text{Dedendum} = 1.157 \times \text{module}.$$

$$\text{Outside diameter of gear blank} = \text{Pitch diameter} + (2 \times \text{module}).$$

The standard modules progress in steps of ¼ from 1 to 4; then in steps of ½ to 7; then in steps of 1. A module m is equal to a diametral pitch of $25.4/m$.

CHAPTER IV

Change-Speed Gears

Three types of mechanism are used for multiplying the torque of the engine in different proportions, in accordance with different requirements. The first of these is the sliding-pinion change-speed gear, which is used on practically all passenger cars; the second is the constant-mesh change-speed gear, in which there are a number of pairs of gears on parallel shafts, which constantly remain in mesh with each other, one gear of each pair being normally free on its shaft, and locked thereto by a positive clutch only when it is desired to engage that particular gear; the third is the electric transmission, which at the present time is used extensively only on large vehicles, such as motor buses and rail cars. With this the energy of the engine is converted into electric energy in a generator coupled to the engine, this electric energy being supplied to electric motors geared to the driving wheels.

Obsolete Types—In the past two other types of change-speed gear have been used for motor vehicle purposes. One is the friction disc transmission, comprising a friction disc usually mounted on the engine crankshaft, with which can be brought into engagement a friction wheel on a shaft transverse to the crankshaft. By sliding the friction wheel along its shaft to a greater or lesser distance from the axis of the disc, the rate of torque multiplication can be varied. One reason for the abandonment of this gear was that with the great increase in the engine power of passenger cars, it had to be made very bulky in order to get sufficient transmission capacity. Besides, inexperienced drivers sometimes wore flat spots on the wheel by continuing to run the engine when the car was stalled, whereupon the operation became very unpleasant.

The other type referred to is the planetary gear. This was used as a two-speed-and-reverse gear on all of the cars built by the Ford Motor Company up to 1927 (some ten million), but it is rather inefficient on low gear, becomes noisy

quickly, and leads to great complication when it is desired to develop a three-speed gear.

History of the Sliding Gear—The transmission of the power from an internal combustion engine of relatively little flexibility to the wheels of a road vehicle in such a way as to make it possible to meet the requirement of wide variability of speed and torque, always has been a knotty problem. Daimler and Benz, the pioneers of the gasoline automobile, used belts and stepped pulleys in their earliest designs. The Daimler engine was taken up in France by the firm of Panhard & Levassor, and after a few experiments with belts, the engineer of the firm, M. Levassor, introduced the sliding-pinion change-speed gear in conjunction with a leather-faced cone clutch. The idea of meshing toothed gears by shifting them axially was at first ridiculed as crude and unmechanical, but after the gear had undergone a number of important refinements and modifications it proved more satisfactory than any competing system, for the lighter types of vehicle at least, and today it is in almost universal use on passenger cars and in extensive use on other types of vehicle and on tractors.

Levassor's change gear is illustrated in Fig. 1. It consisted of two parallel shafts mounted in bearings in an aluminum gear box. The first of these, known as the primary shaft, was in driving connection with the clutch. This shaft was squared and carried a set of three toothed gears or pinions, whose common hub had a square hole broached through it to make a sliding fit with the square shaft. On the secondary shaft were carried three other toothed gears, each of such a diameter as to properly mesh with one of the gears on the primary shaft. The gears on both shafts were so spaced that by shifting the primary set, corresponding gears on the two shafts could be brought into mesh successively without interference from the other gears. Shifting of the sliding set was accomplished by means of a hand lever located convenient to the operator, and a suitable connecting linkage. The secondary shaft at its rear end carried a bevel pinion meshing with a bevel gear on a cross shaft or jackshaft, from which the power was transmitted to the rear wheels by side chains.

One disadvantage of Levassor's gearset was that the power was transmitted through a pair of toothed gears—with consequent power loss, noise and wear—even at high car speeds, when there was absolutely no occasion for it, since the speed was not changed by the gearing. This objection was overcome in a change gear brought out some years later by

Louis Renault, which differed from Levassor's in that the gears of the two shafts were rolled (instead of being slid) into mesh. The primary shaft of this gearset was in two parts, the forward or driving, and the rear or driven part, the forward end of the latter being piloted inside the former. The secondary shaft served as a countershaft through which the motion was transmitted for low and intermediate speed and for reverse. For high speed the two parts of the primary shaft were locked together by means of clutch jaws formed integral with a gear on each of them, which could be slid into engagement. This gave the so-called direct drive, the power being carried directly through the gearset without

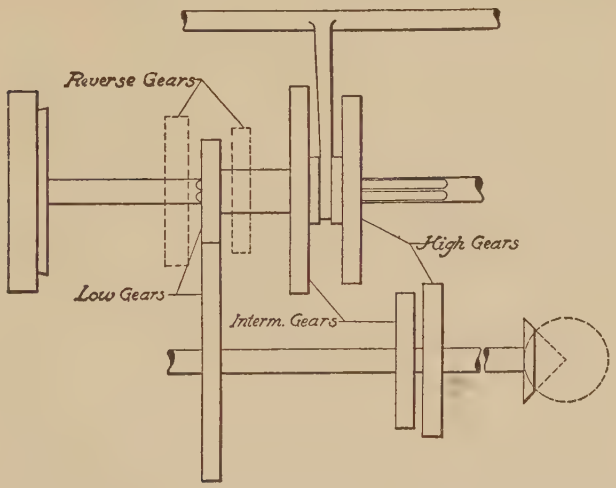


FIG. 1.—SKETCH OF LEVASSOR'S SLIDING-PINION CHANGE-SPEED GEAR.

being transmitted through toothed gears. Soon the direct drive feature was incorporated also in the Levassor type of sliding gear, as shown in Fig. 2. This gear, which became known as the three-speed progressive-type of sliding gear with direct-drive on high, was used extensively for many years, but is now obsolete.

As the speeds of automobiles increased, it became customary to fit change gears giving four forward changes of gear and one reverse, to enable the operator to run his engine near its speed of highest efficiency under practically all operating conditions. Now, a four-speed progressive-type transmission comes out exceedingly long. Not only does this lead

to a bulky and heavy gearbox, but the shafts, being relatively long, will yield excessively under tooth loads, with the result that the transmission is noisy and inefficient. The great length with this construction is due mainly to the fact that the gears on each shaft must be spaced relatively far apart to avoid interference. This difficulty was first overcome by Wilhelm Maybach, engineer of the Daimler Motor Company of Cannstatt, Germany, who used two sliding sets in a non-direct-drive type of transmission. This principle was later applied to the direct-drive type and proved so popular that at present it is used exclusively for both three- and four-speed sliding gear transmissions.

Three-speed-and-reverse gears usually have two sliding sets; four-speed gears, three. The several sliding sets are

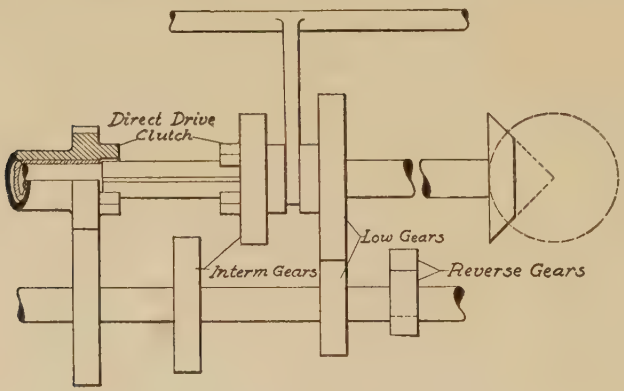


Fig. 2.—Sliding-Gear Transmission with Direct Drive (Progressive Type).

operated by means of a single lever. This type of change gear is known as the selective sliding gear type. It has the advantage over the progressive type that the driver may change from any one gear to any other without passing through intermediate gears. A sketch of a four-speed selective sliding gear is shown in Fig. 3, from which the compactness of this type of transmission is apparent.

Materials for Transmission Gears—Steels of high quality must be used for the gears of the transmission, for the stresses on them are exceptionally high. The teeth of the gears must not only be able to withstand the bending stresses due to the tooth pressure, but their surfaces must be sufficiently hard so that no pitting will occur under the very high specific

pressures; yet they must be not so brittle that they will scale or chip under the impact of clashing.

Two general types of steel are employed for the gears—case-hardening steels and oil-hardening steels. The former are characterized by a low carbon content, usually about 0.20 per cent, while the latter have a medium carbon content of 0.45 or 0.50 per cent. In each group the cheapest steel is the plain carbon steel, the S. A. E. steel No. 1020, for instance in the former, and the S. A. E. steel No. 1045 in the latter. But plain carbon steels are no longer used for transmission gears, at least not to any extent, because they have insufficient strength.

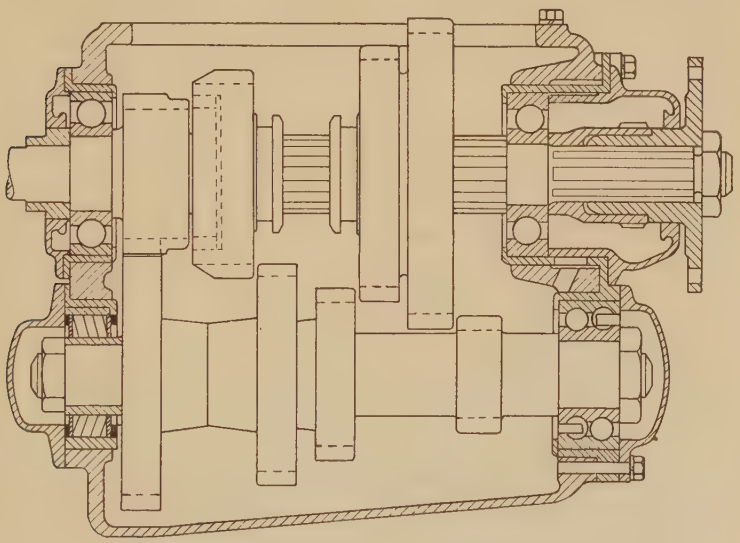


Fig. 3.—Four-Speed Selective Sliding-Gear Transmission.

Case-Hardening Steels—The most popular case-hardening steel for this purpose is the S. A. E. steel No. 2320, which has the following composition (according to specifications):

C, 0.15–0.25; Mn, 0.50–0.80; P, 0.04 max.;
S, 0.045 max.; Ni, 3.25–3.75.

The heat treatment for this steel when used for clash gears is as follows: After the gears are cut, carburize at 1600°–1650° F. and cool in box; then reheat to 1375°–1425° F. and quench in oil; draw at 375° F. With this treatment a scleroscope hardness of about 85 is obtained.

There are several other alloy case-hardening steels which are used to a lesser extent for transmission gears. The 2512, also known as the 5 per cent nickel, case-hardening steel, differs from the 2320 in having a lower carbon content (0.17 per cent max.) and a higher nickel content (5 per cent nominal instead of 3.5). This steel is quite expensive and is used only by one or two manufacturers of high-priced cars. It gives an exceptionally tough core. In order to reduce the distortion following the carburizing process, it is well to give this steel a normalizing treatment or a quench and draw, which will also improve its machining qualities.

Similar to the No. 2320 is the No. 3120, a nickel-chromium steel. Instead of 3.5 per cent nickel nominal, it contains only 1.00–1.50 per cent nickel, but in addition 0.45–0.75 per cent chromium. For the same heat treatment the tensile strength and yield point are slightly less than for the 3.5 per cent nickel steel.

A further case-hardening steel is the No. 6120, a chrome-vanadium steel with the following composition:

C, 0.15–0.25; Mn, 0.50–0.80; P, 0.04 max.;
S, 0.04 max.; Cr, 0.80–1.10; Va, 0.15 min.

The heat treatment is the same as for the 3.5 per cent nickel steel No. 2320.

Oil-Hardening Steels—A typical representative of the oil-hardening family of steels is the No. 2345, the composition of which is the same as that of No. 2320 except that the carbon content lies between 0.40 and 0.50 per cent. The heat treatment for gears made from this material in the form of forgings is as follows: Normalize at 1625°–1725° F.; heat to 1425°–1475° F. and quench in oil; draw at 400° F. A scleroscope hardness of about 72 will be obtained. With gears of this material, therefore, the surface hardness will not be so great as with case-hardened gears, but, on the other hand, the tensile strength of the material in the core will be materially greater, owing to its higher carbon content.

A similar material is the No. 3245 nickel-chromium steel, which receives the same heat treatment and has slightly better physical properties. Other oil-hardening steels used for gears are the No. 5150, a chromium steel, and the No. 6150, a chromium-vanadium steel, the carbon content in each case being 0.45–0.55, and the chromium content 0.80–1.10, while in the case of the chromium-vanadium steel a vanadium content of 0.15 per cent min. (0.18 per cent desired) is specified. When the gears are made from chromium steel in the form of forgings they are given the following heat treatment:

Normalize at 1600°–1700° F.; reheat to 1450°–1550° F. and quench in oil; draw at 400° F. For the chromium-vanadium steel both the normalizing and reheating temperatures are somewhat higher, the former being specified at 1650°–1750° F. and the latter at 1525°–1625° F.

Comparing oil-hardened with case-hardened steels for clash gears, the former have a lower surface hardness, generally from 430 to 500 Brinell, as compared with from 530 to 600 Brinell. But the oil-hardened steels have a higher tensile strength, and, besides, they distort less during heat treatment.

Cyanide-Hardened Gears—In order to obtain greater surface hardness on gears of medium carbon steel, these gears are now cyanide-hardened by a number of large manufacturers. Sodium cyanide is generally used as the hardening material, instead of the more poisonous potassium cyanide. The chromium steel No. 5150 has been found particularly suitable for this treatment. The treatment given the gears in one case (Dodge Brothers) is as follows: After the gears are cut they are slowly brought up to the temperature of the cyanide bath in an electric furnace and are allowed to "soak" at this temperature for about 20 minutes. Then they are immersed in the cyanide pots for about one minute, and finally, as they come out of the pots, they are quenched in oil.

In the cyaniding operation the cyanide salt is broken up by the heat, and the nascent nitrogen formed by the decomposition combines with the iron to form very hard nitrides, which are the cause of the great surface hardness, steel of medium carbon content being used. Cyaniding produces a gear that combines the great surface-hardness of the case-hardened, with the high core-strength of the ordinary oil-hardened gear.

Nitrided Gears—A special process of surface-hardening steel parts has been developed by the Friedrich Krupp Company of Essen, Germany. It is known as nitriding and is being exploited in this country by several steel companies. The process consists in maintaining the parts treated at a moderate temperature for a considerable time while in the presence of ammonia gas. This gas is decomposed by the heat into its constituents, and the nitrogen liberated combines with the metallic elements of the parts to form nitrides on the surfaces thereof.

Special grades of steel are used for parts to be subjected to this treatment. These steels come in three grades, of the following compositions:

Table I.—Composition of Nitriding Steels

Element	Grade A	Grade D	Grade C
Carbon	0.45	0.32	0.10
Manganese	0.60	0.60	0.60
Silicon	0.25	0.25	0.25
Aluminum	1.00	1.00	1.00
Chromium	1.60	1.70	1.65
Nickel	0.00	1.80	0.00
Sulphur	0.02	0.02	0.02
Phosphorus	0.02	0.02	0.02

There is a dual reason for the inclusion of aluminum in the alloys, as it adds to the hardness of the surface layer and prevents the penetration of the nitrogen beyond a certain depth, thus preventing brittleness of the core. When quenched at 1650° F. (1700° for Grade C) and drawn at 800°, the Grade A shows a maximum tensile strength of 252,000 lbs. per sq. in.; Grade D, 212,500 lbs. and Grade C, 86,000 lbs., which figures reduce to 111,750, 115,750 and 61,000 lbs., respectively, when the drawing takes place at 1400° F., the decrease in tensile strength being substantially in direct proportion to the increase in drawing temperature. When annealed at 1575°, Grade A has a tensile strength of 99,500 lbs. per sq. in. and Grade D, 97,250 lbs.

In preparing parts for the nitriding process it is advisable to first relieve rolling or forging strains by annealing at 1000° F. for about 5 hours. Then the steel is heat-treated to impart the desired properties to the core, and next it is machined. All scale resulting from forging or heat-treating must be removed from the surfaces of the parts to be nitrided. For parts requiring great strength of core, either Grade A or Grade D is used. All sharp edges must be removed from parts, and fairly liberal fillets provided, in order that the extremely hard "case" may not chip off at such points.

The nitriding operation is generally carried out in an electric furnace whose temperature is maintained at 875° F. The parts to be nitrided are placed in a gas-tight box provided with inlet and outlet tubes for the circulation of the ammonia gas. They are subjected to the heat and the ammonia gas for about 90 hours, by which time a case of about 0.031 in. depth is obtained. The case, however, is not of uniform hardness. Results on a test specimen are given by the manufacturers, according to which the Brinell hardness varied from 1002 at the surface to 378 at 0.030 in. depth.

The great advantage of the nitriding process is that, owing to the comparatively low temperature employed, the parts are not distorted by it. On the other hand, the special steels and the length of time required for carrying through the process make parts treated by it rather expensive.

Gear Ratios—In American passenger-car practice, where comparatively powerful engines are the rule, three gear changes have been found to meet the requirements of flexibility fairly well, and up to the end of 1927 the three-speed gear was used almost exclusively. In Europe, on the other hand, where passenger cars are fitted with much smaller engines, four-speed gears are extensively used. Motor trucks of the heavier types also have four-speed gears, both here and abroad, and some even have gears giving seven forward speeds. On farm tractors two and three forward speeds are used with about equal frequency, while in the rail car field, where practice is not yet fully crystallized, either three or four forward speeds are provided.

In the case of passenger cars for the American market, it is found that to meet requirements with respect to acceleration, a piston displacement under gas pressure of 33 cu. in. per ton moved per foot traveled on high gear is required (assuming an engine of normal specific output with a compression ratio of about 5 to 1). Also, in order that the car may be able to negotiate the most difficult roads under full load, the piston displacement under gas pressure must be about 105 cu. in. per ton moved per foot traveled in low gear, this being sufficient in most cases to just slip the driving wheels on dry pavement. Hence the low gear reduction must be $105/33 = 3.18$, and this is about the ratio usually employed.

Geometrical Series of Ratios—As regards the ratio for the intermediate gear, from one point of view the most logical plan is to so proportion the various gears that they form a geometrical series. That is, the ratio for the intermediate gear should be $\sqrt{3.18} = 1.78$. While this rule of a geometrical series is substantially followed, generally the ratio between the first and second speeds (low and intermediate) is made somewhat greater than the ratio between the second and third. The reason for this is that while the low speed must be low enough to meet all emergencies, it is desirable to lessen the step between the second and third, as this makes it easier to make this shift, which must often be made in traffic, and it also gives a somewhat higher car speed on second gear, which is advantageous for obvious reasons.

When four gear changes are provided, as on the heavier

trucks, the individual steps can be made somewhat smaller and the total range be still considerably greater. Average ratios for a four-speed truck transmission are as follows:

$$1—1.72—1.72^2—1.72^3,$$
$$1—1.72—2.92—5.07.$$

Of course, it is not always possible to get a desired ratio exactly, because, for reasons of strength, comparatively coarse teeth must be used, which means a relatively small number of teeth, and the addition of one tooth to one gear and the subtraction of one from the mating gear results in a considerable change in the ratio. Gear combinations to give the

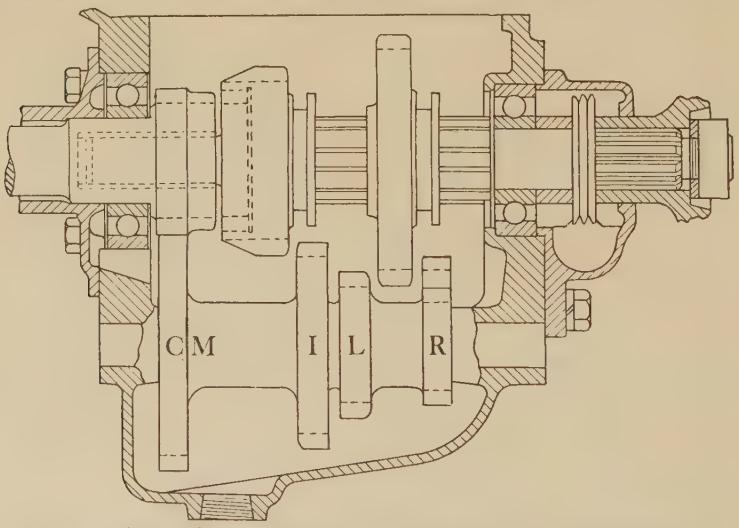


Fig. 4.—Three-Speed Selective Sliding-Gear Transmission.

desired ratios as nearly as possible must be worked out by trial and error.

The ratio of the reverse gear is generally made somewhat greater than that of the low speed forward—as low as the design permits—because this assures greater safety in backing.

Arrangements of Gears—Referring to Fig. 4 which is a vertical section through a modern three-speed transmission, it will be seen that the shaft entering the gear housing from the left—and which in unit power plants is the clutch shaft—carries a pinion which meshes with a gear on the secondary shaft below it. These two gears remain in mesh continuously, while the rest of the gears are shifted into mesh, two and

two, when it is desired to use them. It will be noticed that the gear on the secondary shaft is of nearly twice the diameter as the driving pinion on the clutch shaft, hence the secondary shaft runs at all times at a little more than half the speed of the engine. There is an alternate construction in which the constant mesh set of gears is located at the rear end of the gearbox, but this is subject to the disadvantage that when the direct drive is in operation, which it is a large proportion of the time the car is in use, the secondary shaft runs at nearly twice engine speed and the pitchline velocity of the constant-mesh gears is nearly twice as great. When this arrangement was used, in the early years of the industry, it was customary to disengage the so-called "constant-mesh" gears when engaging the direct drive.

Tooth Numbers—In laying out the gear trains it is to be remembered that gears with less than a certain number of teeth do not give a smooth tooth action. Fifteen is the smallest number of teeth which should be used in a transmission gear, and this should preferably be used only in the low speed pinion, in the case of which smooth action and silent operation are not so essential. For the pinion of the constant-mesh set it is advisable not to choose less than 16 teeth.

One of the principal dimensions in a transmission gear is the distance between shaft axes. If we study the Lewis formula,

$$W = S\,p\,f\,y,$$

we see that the tangential force which a certain gear is capable of transmitting is proportional to the product of two of its linear dimensions, its circular pitch and its face. With definite numbers of teeth in the driving and driven gears, the circular pitch will be proportional to the center distance, and the face width may also be considered proportional thereto, for with increase in pitch diameter of a gear the face can be widened.

The torque capacity of a gear is equal to the tangential force which it is capable of transmitting, multiplied by its pitch radius, which latter, for a certain number of teeth, also is proportional to the center distance. Hence we may say that the torque capacity of a pair of gears is proportional to the cube of its center distance. An investigation of numerous transmissions in regular production shows that for passenger car (light-duty) transmissions

$$d = \sqrt[3]{T/4} \text{ in.},$$

and for truck and bus (heavy-duty) transmissions

$$d = \sqrt[3]{T/2} \text{ in.},$$

where T is the maximum torque the engine is capable of, in lb.-ft. Transmissions for motor trucks and buses are made much larger than transmissions for passenger cars, for the reason that in the first-mentioned classes of vehicle the displacement factor (piston sweep in cubic inches per ton per foot of movement) is much less, and as a result the lower gears of the transmission must be used a greater proportion of the time.

It was previously shown that in a three speed gear the low-speed ratio is usually about 3.18 to 1 and the ratio of the intermediate gear, 1.78 to 1. Usually about the same ratio is used for the low speed pair as for the constant mesh pair, while the intermediate pair of gears are made with a 1 to 1 ratio. These proportions, of course, can be slightly varied.

Choice of Pitches—If we decide on a minimum number of teeth of 16 and on a ratio of 1.78 for the constant-mesh set, then the ranges of center distance and torque for the different pitches are at once fixed. If p is the diametral pitch, the pitch diameter of the pinion will be $16/p$ and that of the gear, $1.78 \times 16/p$, and since the center distance is equal to one-half the sum of the pitch diameters, it is

$$(2.78 \times 16)/2p = 22.2/p.$$

Equating this to the center distance in terms of the torque (found above) we have

$$22.2/p = \sqrt{T/4},$$

from which it follows that

$$T = 44{,}000/p^3,$$

T being the smallest engine torque for which the particular diametral pitch should be used. Equations for T and d give the results given in Table II.

Table II.—Diametral Pitches for Light-Duty Transmissions

Diametral Pitch	Least Center Distance	Least Engine Torque
8	2.78 in.	86 lb.-ft.
7	3.18 in.	128 lb.-ft.
6	3.71 in.	203 lb.-ft.

In the larger trucks and buses having four-speed gears the constant-mesh ratio, as previously shown, is in the neighborhood of 1.72. Moreover, the number of teeth in the constant-mesh pinion would hardly be made less than 18, hence the center distance is

$$(2.72 \times 18)/2p = 24.5/p,$$

and since the torque is related to the center distance by the formula

$$d = \sqrt[3]{T/2},$$

we get for the least torque for any particular diametral pitch,

$$T = 29{,}400/p^3.$$

These two equations give the following results for different diametral pitches:

Table III.—Diametral Pitches for Heavy-Duty Transmissions

Diametral Pitch	Least Center Distance	Least Engine Torque
6	4.08 in.	136 lb.-ft.
5	4.97 in.	235 lb.-ft.
4½	5.45 in.	322 lb.-ft.

In order to simplify shop equipment, a given pitch is sometimes used for a wider range of capacities than would seem logical on other grounds.

Allowable Tooth Stresses—In the application of the Lewis formula to the teeth of three-speed passenger-car transmissions the following values for the stress S can be figured with:

Const. mesh set	Intermediate set	Low speed set
40,000 lbs.	45,000 lbs.	70,000 lbs

Similarly, for four-speed truck or high-duty transmissions the following stresses may be allowed:

Const.-mesh set	Third-speed set	Second-speed set	Low-speed set
16,000 lbs	18,000 lbs.	24,000 lbs	32,000 lbs.

These figures are low enough to be used with any of the alloy steels mentioned in the earlier part of this chapter, though the factor of safety, of course, will be greater for the high carbon steels. The S. A. E. physical property charts indicate that the case hardening alloy steels, such as Nos.

2320 and 3120, when given the heat treatment described in the foregoing, that is, when drawn to 450°, have a tensile strength of approximately 200,000 lbs. per sq. in., while the oil hardening steels, such as 2350, when heat- treated as described, have a tensile strength of about 250,000 lbs. per sq. in. Therefore, although, the allowable stress given may seem quite high, the factor of safety is entirely adequate.

In the use of the foregoing values in the Lewis formula, the full width of all teeth can be figured with, as the chamfering does not materially reduce the section at the root of the tooth, where the stress is a maximum.

So far as safety from tooth breakage is concerned, practically the same tooth stresses could be allowed in the gears of heavy-duty transmissions as used on vehicles (such as heavy trucks and buses) which operate on the intermediate speeds a good deal of the time. However, if the stresses are determined from the dimensions of the gears of actual heavy-duty transmissions, they are found to be only about half as high as in passenger-car transmissions. The reason is that, in order to obtain adequate life under heavy duty conditions, the compressive stresses must be kept down, so as to reduce the rate of tooth wear. The compressive stress, therefore, is the determining factor and it is sufficient to calculate the gears of heavy-duty transmissions on this basis alone.

Allowance for Chamfering—In calculating the maximum compressive stress on the faces of the gear teeth, in the case of sliding gears only the effective width of face (total width minus width removed by chamfering) should be figured with. The reduction in width due to chamfering varies somewhat with the form of the chamfer and it also varies with the pitch of the teeth. The following allowances for the reduction in width due to chamfering are representative of good practice.

Diametral pitch	8	7	6	5	4½
Chamfering width	⅛	5/32	3/16	7/32	¼

Permissible Compressive Stresses—An analysis of numerous modern passenger car or light-duty transmissions shows that the maximum static compression stresses allowed with the different pairs of gears are about as follows (lbs. per sq. in.):

Constant-mesh set	Intermediate gears	Low-speed gears
200,000	250,000	300,000

Similarly, for four-speed truck and bus or high-duty transmissions the compression stresses allowed are about as follows (lbs. per sq. in.):

Const.-mesh set	Third-speed	Second-speed	First-speed
137,500	145,000	160,000	215,000

Greater static compressive stresses can be allowed for the lower speed gears because their pitchline velocities are lower and therefore the load increment due to dynamic forces is less. The compressive stresses in the teeth of passenger-car transmissions will be seen to be exceedingly high, especially when it is considered that the figures given are still further materially increased by the increment loads. That such high stresses are allowable is due solely to the fact that American passenger cars are so highly powered that they can take practically everything on high gear, and the lower gears are operated for short periods at a time only. Perry L. Tenney (S. A. E. Journal, Oct., 1925), found that one hour's operation of the intermediate gears of a three-speed transmission on the dynamometer stand under full engine load, is equal to 3000 miles operation in a passenger car. In the heavier service of stage coaches in mountainous or sandy districts it is equal to only 1000 miles of service. European passenger cars, which are less highly powered than American makes, carry relatively larger transmissions, as these cars are being operated much more on their gears.

Sample Calculation—We will now illustrate the application of the Lewis formula to the calculation of a three-speed transmission for a car equipped with a 225-cu. in. engine, which may be assumed to develop a maximum torque of 145 lb.-ft. The center distance for the shafts according to the equation given should be about

$$\sqrt[3]{145/4} = 3.3 \text{ in.}$$

The proper pitch to use is 7 diametral pitch (stub teeth). With a center distance of 3.285 in. a total of 46 teeth can be used in the pinion and gear together, and since a ratio of about 1.75 is required for the constant mesh set, the number of teeth in the pinion should be

$$46/(1 + 1.75) = 16.72,$$

the nearest whole number to which is 17. We may therefore lay out the following gear combination:

Constant mesh set	Intermediate gears	Low-speed gears
17 : 29	23 : 23	16 : 30

This combination gives a somewhat greater ratio between intermediate and low than between high and intermediate, which, as previously pointed out, is a desirable feature.

The pitch radius of the constant mesh pinion will be

$$17/14 = 1.214 \text{ in.} = 0.101 \text{ ft.}$$

Hence, the tangential force on the pitch circle of the pinion will be

$$145/0.101 = 1440 \text{ lbs.}$$

The circular pitch of a 7-diametral-pitch gear is 0.449 in., and if we use the American Standard stub tooth, the value of the Lewis constant for 17 teeth is 0.110. Hence, inserting in the Lewis equation, we find for the width of teeth required

$$f = \frac{1440}{40{,}000 \times 0.449 \times 0.110} = 0.73 \text{ in.}$$

The torque on the secondary shaft evidently is

$$145 \times (29/17) = 248 \text{ lb.-ft.}$$

The pitch radius of the intermediate pinion is

$$23/14 = 1.64 \text{ in.} = 0.137 \text{ ft.}$$

Therefore the tangential force on the pitch circle of the intermediate pinion is

$$248/0.137 = 1810 \text{ lbs.}$$

The Lewis factor y for 23 teeth and an American Standard stub tooth is 0.124, hence the required width of face is

$$f = \frac{1810}{45{,}000 \times 0.449 \times 0.124} = 0.725 \text{ in.}$$

The pitch radius of the low-speed pinion is

$$16/14 = 1.143 \text{ in.} = 0.0954 \text{ ft.}$$

The tangential force on the pitch circle of this pinion is

$$248/0.0954 = 2600 \text{ lbs.}$$

The Lewis constant for 16 teeth and the American Standard stub tooth is 0.105, hence the required width of face is

$$f = \frac{2600}{70{,}000 \times 0.449 \times 0.105} = 0.788 \text{ in.}$$

Calculation on Basis of Compression Stresses—We will now calculate the same transmission on the basis of the compressive stresses. The tangential force on the pitch circle of the 17-tooth constant-mesh pinion we found to be 1440 lbs.

The value of $(1 + \rho) = 2.7$, and the radius R of the driven gear is 2.07 in. Since the allowable compression stress is 200,000 lbs. per sq. in., inserting values in the equation for compressive stress in stub teeth, we get

$$200{,}000 = 3900 \sqrt{\frac{1440 \times 2.7}{2.07 f}}.$$

Hence

$$f = \frac{3900^2 \times 1440 \times 2.7}{2.07 \times 200{,}000^2} = 0.715 \text{ in.}$$

The tangential pressure on the pitchline of the intermediate speed pinion is

$$1440 \ (29/23) = 1810 \text{ lbs.}$$

The value of the expression $1 + \rho$ in this case is 2, and the pitch radius of the larger of the two mating gears is 1.64 in. Since the allowable compressive stress is 250,000 lbs. per sq. in., we have for the required effective width of face,

$$f = \frac{3900^2 \times 1810 \times 2}{1.64 \times 250{,}000^2} = 0.538.$$

Since this gear must have its teeth chamfered at one side and the chamfering allowance for 7 pitch teeth is 5/32 or 0.157 in., the total width of the intermediate gears should be

$$0.538 + 0.157 = 0.695.$$

The tangential force on the low-speed pinion is

$$1440 \ (29/16) = 2600 \text{ lbs.}$$

The value of the expression $1 + \rho$ is 2.87, the pitch radius of the low-speed gear, 2.14 in. and the permissible compressive stress, 300,000 lbs. per sq. in. Hence the effective width of face for the low-speed gears should be

$$f_e = \frac{3900^2 \times 2600 \times 2.87}{2.14 \times 300{,}000^2} = 0.592 \text{ in.}$$

Adding to this the allowance for chamfering, 0.157 in., we get for the required total width of the low-speed set of gears, 0.749 in.

The formula based on the compressive stresses therefore gives substantially the same values as the formula based on bending stresses.

Calculation of Truck Transmission—In order to show the application of the formula to a heavy-duty transmission we will now calculate a transmission to go with an engine capable

of a maximum torque of 250 lb.-ft., such as might be used for a 5-ton truck. This torque capacity is within the range for which 5-pitch stub teeth are ordinarily used. According to the formula previously developed, the center distance should be

$$d = \sqrt[3]{250/2} = 5 \text{ in.},$$

which with 5-pitch teeth would give a total of 50 teeth in a pair of mating gears. This formula is intended to cover both truck and bus transmissions, and since truck engines generally run at somewhat lower speed than bus engines, we are justified in choosing a slightly greater number of teeth and center distance. We will use 52 teeth, giving a center distance of 5.2 in.

In determining the gear combinations we have to proceed partly by trial and error. This being a four-speed gear, the reduction on low should be about 5 to 1. Now, the reduction obtained by the constant mesh gears is usually slightly less than the reduction of the low speed set of gears. If they were both to be the same, each would have to be equal to the square root of 5, or 2.24. Since the total number of teeth of the pair is 52, the number of teeth in the pinion then must be

$$52/(2.24 + 1) = 16,$$

making the number of teeth in the gear 36. However, as already stated, it is desirable to make the reduction of the constant mesh pair slightly less than that of the other, because this improves the running conditions of the former pair, which operate a far greater proportion of the time, and at far higher pitchline velocity than the low-speed gears. We will therefore choose 17 teeth for the constant-mesh pinion and 35 for the gear, and 15 and 37 teeth for the low-speed pinion and gear respectively. This will give a total reduction ratio of

$$(35 \times 37)/(17 \times 15) = 5.07.$$

For the third speed we have to have a total reduction of about 1.72, and as the constant-mesh gears give a reduction of $35/17 = 2.12$, the third speed set of gears must step the speed up again in the ratio of $2.12/1.72 = 1.225$. Hence the number of teeth in the smallest gear, the driven one, must be

$$52/(1.225 + 1) = 23 +.$$

We will therefore select for this pair gears with 29 and 23 teeth, which will give a total reduction ratio of

$$(35 \times 23)/(17 \times 29) = 1.63.$$

For the second-speed pair of gears we find in a similar way gears with 22 and 30 teeth, which give a total reduction ratio of 2.81. Thus the total ratios of the four speeds are 1, 1.63, 2.81 and 5.07. The ratios between succeeding gears are as follows: High to third, 1.63; third to second, 1.72; third to low, 1.80. Thus each succeeding step is a little larger, a desirable condition, as it will facilitate the change from high to third, which will have to be made most frequently. While the number of teeth in the low-speed pinion, 15, is smaller than is desirable in a pinion that is used a good deal, this small number is permissible in this case because the low gear of a four-speed transmission is used in emergencies only, and then only for very short periods.

We will next calculate the required widths of face. The pitch radius of the constant-mesh pinion is 1.7 in. = 0.1415 ft. Hence the tangential force on the pitchline of this pinion under full engine torque is

$$250/0.1415 = 1765 \text{ lbs.}$$

The value of $(1 + \rho)$ is 3.06, the radius of the larger gear of the pair, 3.5 in., and the permissible compression stress, 137,500 lbs. per sq. in. Hence, inserting these values in the formula for face width, we find.

$$f = \frac{3900^2 \times 1765 \times 3.06}{3.5 \times 137{,}500^2} = 1.24 \text{ in. (say 1.25 in.)}$$

In a similar manner we find for the required effective width of the third-speed set of gears

$$f_e = \frac{3900^2 \times 2130 \times 2.26}{2.9 \times 145{,}000^2} = 1.20 \text{ in.}$$

With 5-pitch teeth the allowance for chamfering is 0.22 in., hence the total width of face should be

$$1.20 + 0.22 = 1.42 \text{ (say, } 1\tfrac{7}{16} \text{ in.).}$$

For the second speed pair of gears we have

$$f_e = \frac{3900^2 \times 2810 \times 2.36}{3 \times 160{,}000^2} = 1.31 \text{ in.}$$

and adding the chamfer allowance,

$$1.31 + 0.22 = 1.53 \text{ (say, } 1\tfrac{1}{2} \text{ in.).}$$

For the low-speed pair of gears we have

$$f_e = \frac{3900^2 \times 4120 \times 3.465}{3.7 \times 215{,}000^2} = 1.27 \text{ in.}$$

Adding the chamfering allowance,

$$1.27 + 0.22 = 1.49 \text{ (say, } 1\tfrac{1}{2} \text{ in.)}.$$

Bearing Loads—The shafts of automotive-type transmissions are generally mounted in anti-friction bearings, because the requirements of such bearings with respect to lubrication are more readily met, and their use results in a reduction in frictional losses and in wear. As a result of the reduction in wear the transmission will remain quiet in operation longer. Where it is necessary to keep down the radial dimensions of the bearing, as where one shaft is supported or piloted inside another, or where a cluster gear revolves on a stationary shaft, Hyatt bearings with helically wound flexible rollers are largely used, while bearings mounted in the wall of the case are either of the annular ball or the conical roller type. The spool gear for the reverse usually is fitted with a bronze bushing, as this gear revolves only when the vehicle is being backed. Secondary shaft gear clusters also are mounted on plain bearings on some of the lower-priced cars, but the tendency is toward the adoption of roller bearings in this place, as it is difficult to keep the transmission quiet if plain bushings are used there. The anti-friction bearings are one of the chief items of expense in connection with transmission gears. Sizes must be selected adequate for the loads on them under full engine power, but in order to produce a competitive product the bearings should be no larger than necessary, which calls for an accurate calculation of bearing loads.

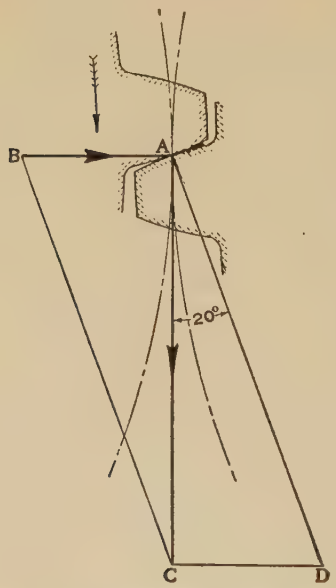


FIG. 5..—COMPOSITION OF GEAR-TOOTH PRESSURE.

In Fig. 5 is shown a diagram of a pair of gear teeth in contact. We will assume that the teeth are of stub form and have a pressure angle of 20°. The tooth pressure is a maximum when the contact is at the pitch point, at which time

the contact surfaces make an angle of 20° with the line of centers. The pressure between the two teeth, which is represented by the line AD, is normal to the contact surfaces. On the other hand, the tangential load on the gear, which is represented by the line AC, is normal to the line of centers and therefore makes an angle of 20° with the tooth pressure AD. In fact, the tooth pressure may be resolved into two components, one, AC, normal to the line of centers and tangential to the pitch circles, which causes the driven gear to turn; the other, AB, in the line of centers, which tends to force the gear shafts apart.

Let T be the torque of the driving pinion and r its pitch radius, then the tangential force at the pitch circle is

$$AC = \frac{T \times 12}{r}.$$

and the tooth pressure is

$$AD = \frac{T \times 12}{r \times \cos 20°}.$$

When calculating the bearing loads for gears that run dry, that is, without lubrication of the tooth contact surfaces, it is customary to take into account the frictional force in the plane of contact, but with gears lubricated like those in an automobile transmission, the friction coefficient is so small that the resulting frictional force is entirely negligible as compared with the tooth pressure, and we will disregard it.

In Fig. 6 A represents the shaft of the driving pinion which has a torque T impressed upon it at some point forward of the bearing, in such a way that no load on the bearing can result from the application of the force creating this torque. The contact surface between the two gear teeth makes an angle of 20° with a tangent to the pitch circles. Now, the teeth with those portions of the gears supporting them may be regarded as two levers. The reaction of lever C on lever B produces a moment $P \times r$ around the axis of primary gear shaft A. The principle that action and reaction are equal and opposite applies to moments the same as it does to forces, and the reaction of the bearing on shaft A tends to turn lever B around the center line of contact D with the same torque but in the opposite direction as the contact pressure P tends to turn the arm around the axis of shaft A. Hence P_1 represents the reaction of the bearing on shaft A and P_2 the pressure of shaft A on the bearing.

Each of the gears is supported on two bearings, these bearings, as a rule, being located on opposite sides of the gear, and the bearing pressure is divided between them in a certain proportion which we shall investigate presently. The constant-mesh pinion in nearly all transmissions is an exception to this rule, since it overhangs its bearing support. From the above we see that the pressure on the bearings supporting any gear is equal to the resultant tooth reaction, and in direction parallel to it. Another thing to be observed is that the pressures on the shafts of two meshing gears due to the pressure between contacting teeth are equal but in opposite directions. This is easily seen, since the pressure of the driving gear teeth against the driven gear teeth is equal to the reaction of the driven gear teeth, but in the opposite direction.

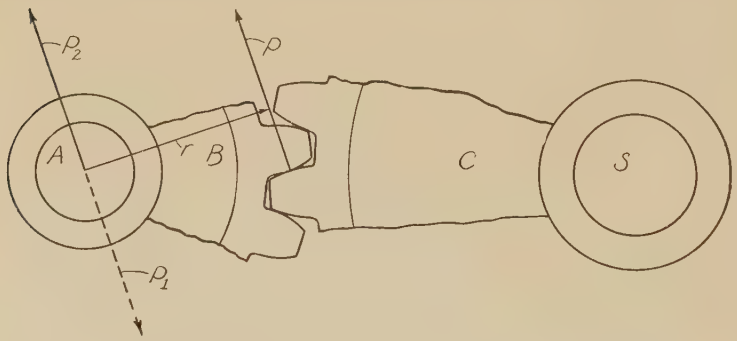


Fig. 6.—Diagram Showing Equality of Tooth Pressure and Bearing Load.

Next it becomes necessary to determine the division of the bearing pressure due to the tooth reaction, between the two bearings supporting any gear. The shaft forms a beam supported at both ends, with a concentrated load at the center of the gear. Referring to Fig. 7, let R_1 and R_2 be the reactions at the supports P, the total bearing load due to one pair of gears; x, the distance of the center of the gear from the center of the left-hand bearing, and y the distance from the right-hand bearing. Then, taking moments around the center of the gear,

$$R_1 x = R_2 y.$$

$$R_1 + R_2 = P$$

$$R_1 = P - R_2$$

$$(P - R_2)x = R_2y$$

$$Px = R_2(x + y)$$

$$R_2 = P\frac{x}{x + y}$$

$$R_1 = P - P\frac{x}{x + y} = P\frac{y}{x + y}.$$

Except when the direct drive is being used, two pairs of gears are in mesh and transmitting power simultaneously, viz., the constant-mesh pair and one of the other pairs. However, the bearing pressures due to these two pairs of gears are not in the same direction and therefore cannot be added together directly, but must be added by means of the paral-

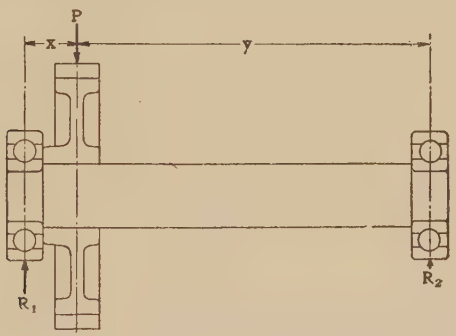


Fig. 7.—Division of Tooth Pressure between Bearings.

lelogram of forces. This may be seen from Fig. 8, which is a front view of the constant-mesh and the intermediate-speed pairs of gears. In this figure, P_1 represents the reaction of the teeth of the constant-mesh gear C on those of the constant-mesh pinion A, and P_2 the pressure of the intermediate-speed pinion D on the intermediate-speed gear B. The loads on the bearings of the primary shaft R are equal and parallel to P_1 and P_2, whereas the loads on the bearings of the secondary shaft are equal and parallel to P_1 and P_2 but oppositely directed. All of these forces make an angle of 20° with the horizontal.

In order to determine the total load on any one of the different bearings corresponding to any particular speed of the gear, we first calculate the bearing load due to one pair of gears, then find the proportion of this on each of its two

bearings; next we determine the bearing load due to the other pair of gears; then find the proportion of this on each bearing, and finally add the two loads on each bearing together by means of the parallelogram of forces, which can be done either graphically or trigonometrically.

We will now carry the calculation through for the four-speed truck transmission of which a lay-out is shown in Fig.

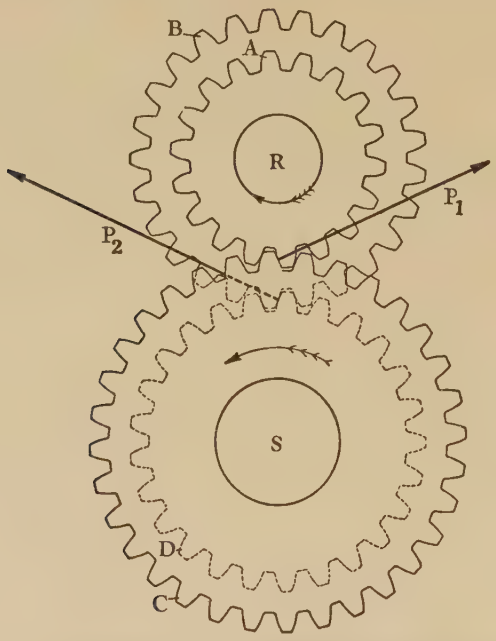


FIG. 8.—CONSTANT-MESH AND INTERMEDIATE-SPEED GEARS (SEEN FROM ENGINE END).

9. This transmission is rated at 250 lb.-ft. and the tangential forces on the pitch circles of the various gears when the full engine power is being transmitted are as follows:

	Lbs.
Constant-mesh set	1764
Third-speed set	2130
Second-speed set	2940
Low-speed set	4120

Since the bearing loads are equal to the tangential forces divided by the cosine of the pressure angle, and the cosine

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of 20° is 0.94, we have for the bearing loads due to these tangential forces

1876 lbs. due to the constant-mesh set of gears,
2266 lbs. due to the third-speed set of gears,
3128 lbs. due to the second-speed set of gears,
4380 lbs. due to the low-speed set of gears.

Now let us assume the third-speed set of gears to be in

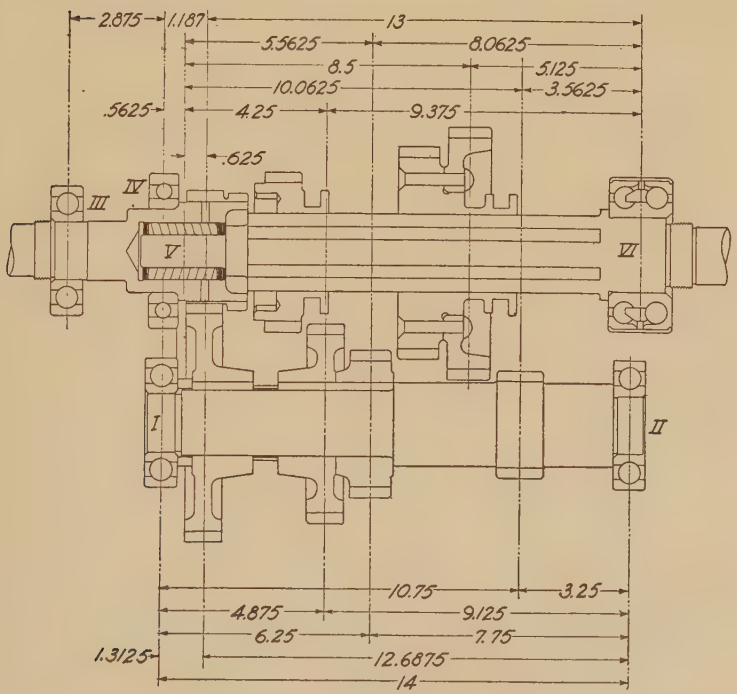


Fig. 9.—Lay-out of Four-Speed Gearset under Calculation.

action. The load on bearing I due to the tooth load on the constant-mesh gears is

$$1876 \times (12.69/14) = 1700 \text{ lbs.},$$

and that on bearing II due to this tooth load is

$$1876 - 1700 = 176 \text{ lbs.}$$

The load on bearing I due to the tooth load on the third-speed set of gears is

$$2266 \times (9.125/14) = 1475 \text{ lbs.}$$

and the load on bearing II due to this tooth load—

$$2266 - 1475 = 791 \text{ lbs.}$$

The forces on bearings I and II for this condition of operation are laid off to scale in the vector diagram Fig. 10, and the resultant of the two forces on each bearing are there found graphically. These resultants can be found also by

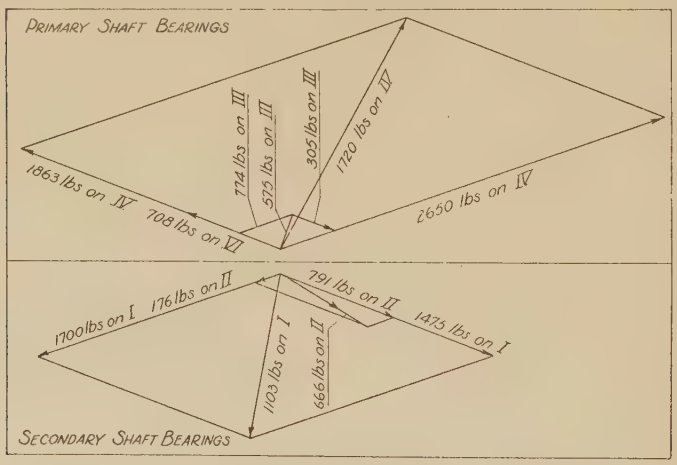


FIG. 10.—BEARING LOADS FOR OPERATION ON THIRD GEAR.

calculation, by making use of the trigonometrical formula for the determination of the length of the third side of an oblique triangle when the other two sides and the included angle are known. This formula is as follows:

$$c = \sqrt{a^2 + b^2 - (2ab \cos \gamma)}.$$

For instance, in the case of bearing I we get for the resultant bearing load

$$\sqrt{1475^2 + 1700^2 - (2 \times 791 \times 1475 \times 0.766)} = 1103 \text{ lbs.}$$

For bearing II we find

$$\sqrt{791^2 + 176^2 - (2 \times 791 \times 176 \times 0.766)} = 666 \text{ lbs.}$$

The directions of the loads on these bearings are as indicated in the diagram.

The load on bearing V due to the tooth load of the third-speed gears is

$$2266 \times (9.375/13.625) = 1558 \text{ lbs.}$$

and the load on bearing VI due to this tooth load is

$$2266 - 1558 = 708 \text{ lbs.}$$

The load on bearing IV due to the tooth load on the third-speed gear is

$$1558 \times (3.44/2.875) = 1863 \text{ lbs.}$$

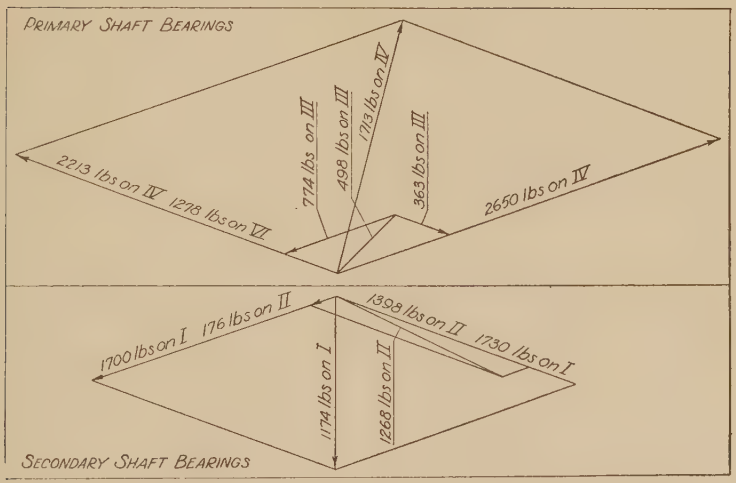


FIG. 11.—BEARING LOADS FOR OPERATION ON SECOND GEAR.

The load on bearing III due to the tooth load on the third-speed set of gears is

$$1863 - 1558 = 305 \text{ lbs.}$$

The load on bearing III is opposite in direction to that on bearing IV.

The load on bearing IV due to the tooth load on the constant-mesh set of gears is

$$1876 \times (4.06/2.875) = 2650 \text{ lbs.}$$

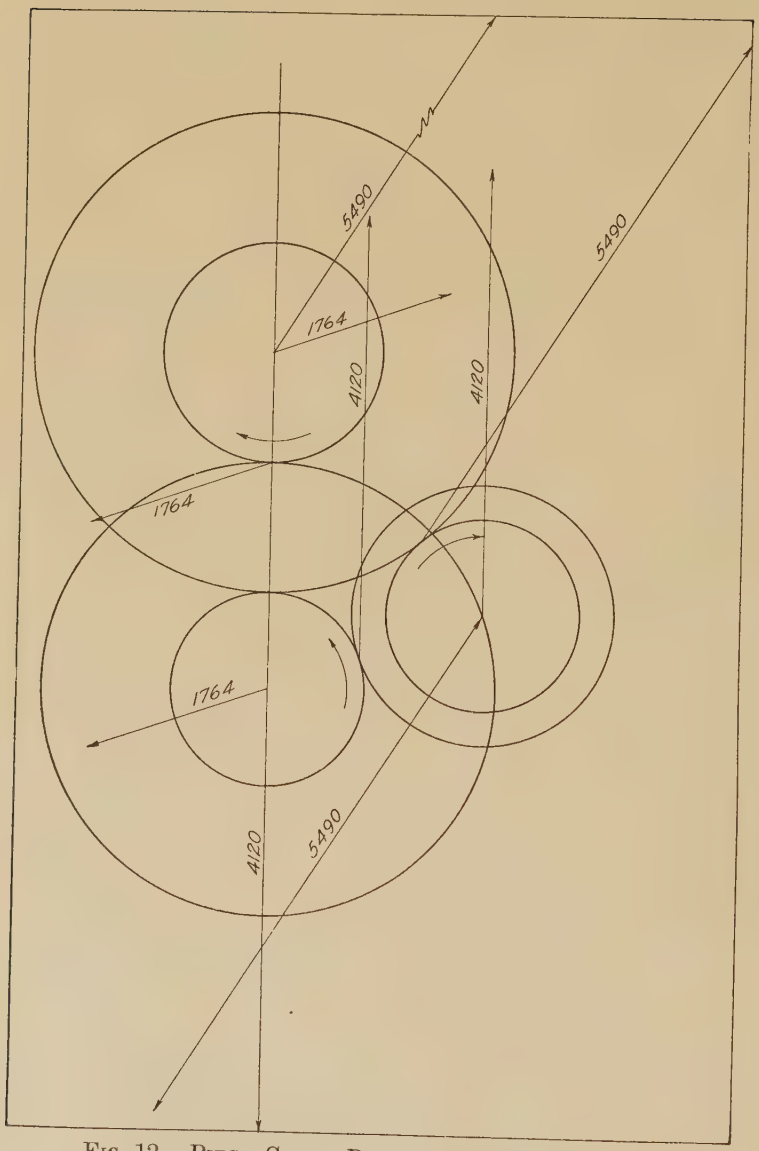


Fig. 12.—Pitch-Circle Diagram of Reverse Gears.

The load on bearing III due to the tooth load on the constant mesh gears is

$$2650 - 1876 = 774 \text{ lbs.}$$

The two loads on each of the bearings III and IV are laid off to scale and in the proper direction in the upper part of Fig. 10 and their resultants are found to be 575 lbs. and 1720 lbs., respectively. The load on bearing VI, 708 lbs., is also laid out on this diagram, while the load on V, 1558 lbs., which is in the same direction as that on bearing VI, is not laid out.

Individual loads on each of the bearings when the second gear is in operation have been calculated in the same way, and they are plotted and graphically combined in the two diagrams of Fig. 11. Loads have been calculated also for low-gear operation, but the load diagram is not reproduced here.

Calculation of the bearing loads due to the reverse gears involves an additional step, in that the directions of the tooth reactions between the reverse idler and the two gears with which it meshes cannot be determined off-hand. It is necessary to lay out the three pitch circles tangent to each other, as has been done in Fig. 12; draw tangents at the points of tangency and then draw lines through the points of tangency making angles of 20° with the tangents, these lines representing the directions of the tooth reactions. The magnitudes of the tooth reactions can be found by the method already explained; after the reactions due to the reverse gears are laid off in the diagram they are divided between the bearings at opposite ends of the shafts carrying the gears, and finally the loads on these bearings due to the reverse gears are combined graphically with the loads due to the constant-mesh gears. This has been done in Fig. 13. The loads on all of the bearings due to all of the different gear combinations under full engine load are tabulated below.

Table IV.—Bearing Loads

Bearing	I	II	III	IV	V	VI
Reverse	2505	2774	1460	4940	2061	3439
First speed	1128	3230	615	1826	1145	3235
Second speed	1174	1268	498	1713	1850	1278
Third speed	1103	666	575	1720	1558	708

Bearing Sizes—It will be seen from Table IV that the greatest bearing loads occur when running in low gear and reverse—the two gear combinations that are used the smallest part of the time. Now, the limiting load which a bearing can carry safely depends on how long that load must be carried. It can be easily seen that if a bearing were rated at the maximum load it is capable of carrying indefinitely, and the bearings were then used in a place where this capacity load occurs only, say, one per cent of the time, it would be much larger than necessary. The calculated bearing loads, moreover, occur only when full engine power is being transmitted, and this happens relatively much less frequently on the lower than on the higher gears.

In three-speed and four-speed transmissions, since the different gear ratios always bear substantially the same proportions to each other, the calculated bearing loads due to operation on the different gears always are more or less proportional, for which reason it has been found practical to simplify the process of determining the necessary bearing sizes by basing it on the calculated loads for second gear. In heavy-duty transmissions the lower gears are used a much larger proportion of the time than in light-duty transmissions, and this can be allowed for by the use of suitable multiplying factors. The New Departure Manufacturing Company, manufacturer of ball bearings, uses bearings of rated capacities substantially equal to the calculated bearing loads in second gear in light-duty transmissions, and uses multiplying factors of about 1.5 for heavy-duty transmissions. In the case of the four-speed transmission for which bearing loads were calculated in the foregoing this method leads to the results tabulated below:

Bearing	I	II	III	IV	V	VI
Second-speed load	1174	1268	498	1713	1850	1278
Required factor	1.5	1.5	1.5	1.3–1.5	1.3–1.5	1.5–2
R.P.M.	600	600	1200	1200	700	500
Required capacity	1760	1900	747	2230	2405	1915
				2570	2775	2555
Bearing	307	307	307	310	404	308
Bearing capacity	1990	1990	1465	2340	2670	2540
Capacity of load	1.68	1.57	2.94	1.37	1.44	1.99

Considerations of ease of production sometimes also affect the selection of bearing sizes. For instance, the maximum load on bearing II is always much greater than that on bearing I, still these bearings are generally made of the same

size because it facilitates machining of the bearing mountings. (In second gear, of course, the loads on the two bearings are nearly alike.) In the case of bearing III the load is generally so small that the bearing size is determined more by the size

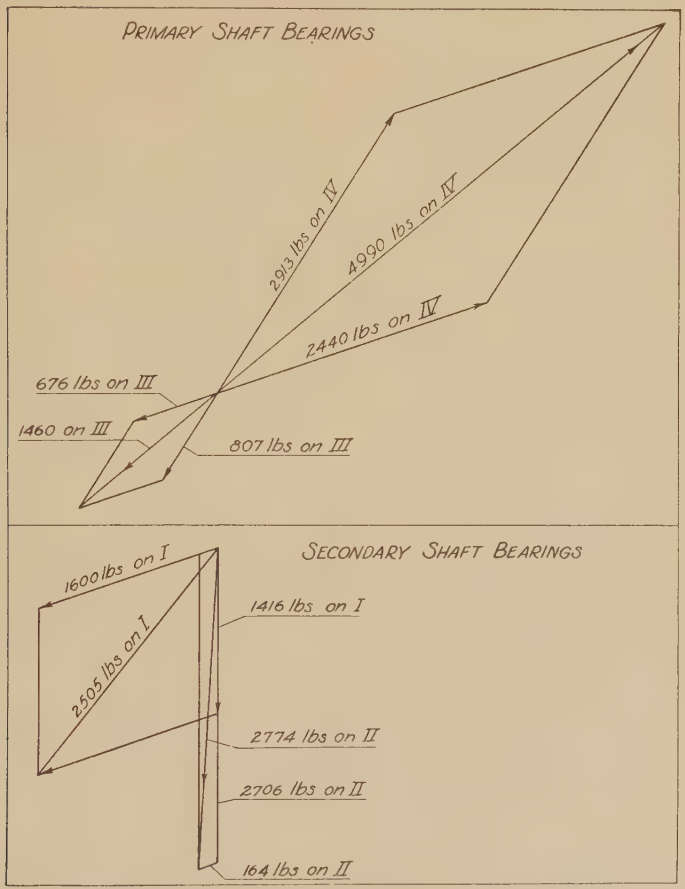


FIG. 13.—BEARING LOADS FOR OPERATION IN REVERSE.

of the clutch shaft on which it is mounted than by the loads it has to sustain.

The load on bearing VI is affected to quite an extent by parts immediately back of the transmission. For instance, if a transmission brake is fitted the maximum load on this bearing is almost certain to be due to that, and this can be

allowed for by using a bearing one size larger than would be used if there were no such brake. This method may seem crude, but there is no really practical method of accurately calculating the bearing loads due to this cause.

Shaft Dimensions—The diameters of shafts for the transmission of power in most instances are calculated on the basis of the torque to be transmitted and the resistance to shear of the shaft material. In the case of automobile and similar transmissions, however, one of the chief requirements is silence of operation of the gears, which is dependent to a large extent upon the rigidity of the shafts and their mountings. The calculation of the necessary shaft diameters therefore is based upon the gear tooth load and the resistance of the shaft to deflection thereby. The deflection generally is a maximum when the pair of gears located most nearly midway between supports is in action, and when this pair of gears transmits the full engine load the deflection should be within the limits of 0.003 and 0.005 in. The flexure of a shaft supported on bearings at its ends and carrying a concentrated load between bearings may be calculated by means of the following equation

$$y = \frac{Pl^3}{8{,}800{,}000\, d^4}\,(2x^4 + 2x^2 - 4x^3),$$

where P is the load on the shaft in pounds; l, the length of the shaft between bearing centers, in inches; d, the diameter of the shaft, in inches; and x, the ratio of the distance of the load from the farthest support to the distance between supports.

Applying this equation to the secondary shaft of the four-speed gearbox calculated in the foregoing, in which the flexure is evidently a maximum when the second-speed gears are in operation, we have

$$\begin{aligned} P &= 2810 \text{ lbs.} \\ l &= 14 \text{ in.} \\ x &= 7.75/14 = 0.55 \\ 2x^4 + 2x^3 - 4x^2 &= 0.125. \end{aligned}$$

If we decide to allow a flexure of 0.005 in. then

$$0.005 = \frac{2810 \times 14^3}{8{,}800{,}000\, d^4} \times 0.125$$

and

$$d = \sqrt[4]{\frac{2{,}810 \times 14^3 \times 0.125}{8{,}800{,}000 \times 0.005}} = 2.16 \text{ in.}$$

The secondary shaft is often made with a somewhat larger diameter at the middle and near the rear end, where the greatest bending moments occur, because of the location there of the gears carrying the highest tooth loads, as this gives the greatest rigidity with the least amount of material.

The limit of deflection for the primary shaft should be the same as that for the secondary shaft, and since its tooth loads are the same and the span between supports is nearly the same, the required diameter will figure out nearly the same as for the secondary shaft. In the case of a splined shaft the mean between the inside and outside diameters can be taken as the effective diameter.

Ball Bearing Mountings.—In mounting ball bearings in the walls of cast aluminum gear cases, use is often made of steel mounting sleeves of somewhat greater width than the ball races, to reduce the unit pressure on the aluminum. The inner race of a radial ball bearing should always be forced onto the shaft under moderate pressure and should be securely clamped between a substantial shoulder on the shaft and a nut which is locked by some reliable means. Of the outer races on a single shaft not more than one should be firmly secured in a lengthwise direction, as otherwise there is danger of subjecting the bearings to undue end thrust. In order to secure the outer race endwise in a bearing hub of little greater length than the race itself, it is customary to insert a bushing in the bore for the mounting with an inward flange at its inner and an outward flange at its outer end. A flanged cap is then inserted into the bushing from the outer end, pressing against the outer race, and bushing and cap are secured in place by cap screws screwed into the wall of the housing.

Transmission Drive Shaft Mounting—The transmission drive shaft through which the power is carried into the gear-case, is mounted differently according to whether the transmission is of the unit type (that is, is to be combined with the engine in a unit powerplant) or is a separate unit. In the former case this shaft is identical with the clutch shaft and has a pilot bearing within the end of the crankshaft or the flywheel web, and another bearing in the forward wall of the transmission case. This latter bearing is so mounted that it will take thrust as well as radial loads. If it is an annular type ball bearing, as is usually the case, its two races are fixed against axial motion on the shaft and in the housing respectively. A light thrust load results from the engagement and disengagement of the high-speed clutch.

If the transmission is a separate unit the drive shaft is

mounted in two bearings, and these must not be mounted too close together, as the load on the inner one of the two—which is the greatest—increases as this distance decreases. One of the two bearings is arranged to take the end thrust while the other has its outer race mounted so it is free to move endwise. A transmission drive shaft with the constant-mesh pinion and its ball bearing is shown in Fig. 14.

Squared and Splined Shafts—In the earliest sliding gear transmissions the shafts on which the gears slide were squared, round shafts of appropriate diameter being milled down to a section whose dimension over the flats was equal to 0.8 the diameter of the original round section. These squared shafts are still used occasionally, but it is much more common to use splined shafts—four-splined for the smaller and six-splined for the larger sizes. These splined sections have been standardized by the S. A. E. and details will be found in the

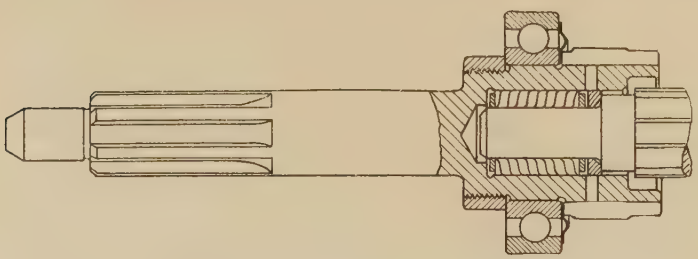


FIG. 14.—CLUTCH SHAFT WITH CONSTANT-MESH PINION.

Appendix. With a squared section there is a tendency for the gear to wedge on the shaft and slide with difficulty after some wear has taken place, and it is also more difficult to ensure concentricity of the gear pitch circle with the shaft axis in production. The splined shafts are finished by grinding after hardening. By grinding out the bores of the gears after hardening and also grinding the bottoms of the spline grooves on the shafts, accurate concentric mounting of the gears is assured. The splined hole in the gear is produced in the first place by broaching.

Direct-Drive Clutch—Two types of direct-drive clutch have been used, viz., the jaw type and the internal-external-gear type. The latter now predominates, undoubtedly because it affords certain advantages from the manufacturing standpoint. One of the members of the latter type of clutch is formed by the constant-mesh pinion, the face of which is made somewhat wider than that of the gear with which it

meshes. Sometimes the height of that portion of the teeth which serves for clutching purposes has to be reduced by turning off the top portion, because of the limited diameter of the intermediate speed gear, on the inside of the rim of which are the internal gear teeth forming the other clutch member. An incidental advantage of the internal-external type of clutch is that it permits of slightly reducing the over-all length of the transmission. On the other hand, this type tends to reduce the length of bearing of the intermediate gear on the splined shaft, which is a disadvantage. The teeth of the internal gear member of the clutch are cut in a gear shaper. The pitch, of course, is fixed by that of the constant-

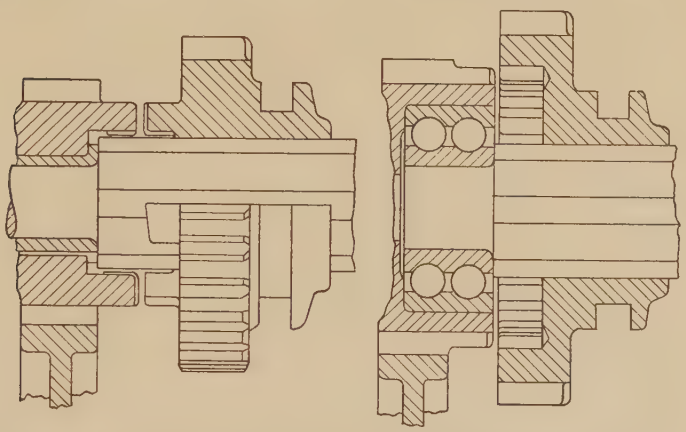


FIG. 15.—JAW AND GEAR-TOOTH-TYPE OF CLUTCH FOR DIRECT DRIVE.

mesh pinion and the width of face can be made about one-third that of the intermediate gear.

Where the jaw-type of clutch is used, four jaws are generally provided, each extending over slightly less than one-eighth the circumference. The outer faces of the jaws have a slight backward rake, which facilitates the engagement, and the contacting surfaces may also be given a slight rake, which tends to prevent automatic disengagement of the clutch as a result of wear of these surfaces. The two types of clutch are illustrated in Fig. 15.

All of the sliding gears are provided with hub extensions with grooves into which the shifter yokes engage. These grooves are generally about square in section, their width and depth varying from 5/16 in. in a 100 lb.-ft. transmission to ½ in. for one of 250 lb.-ft.

Secondary Shaft Assemblies—There are two general types of secondary shaft assembly. The gears may be formed integral with or secured by keying or splining to a shaft which revolves in bearings mounted in the transmission case (Fig. 16), or the gear assembly may form a unit which is bored out and revolves on a stationary shaft or stud rigidly supported in the gear case (Fig. 17). If any of the gears are formed integral with a revolving shaft it is usually only the low speed and reverse pinions, the others being made in the form of separate parts, for convenience in production. So-called cluster gears, however, are often all made in a single piece, in which case it is necessary that some of the gears at least must have their teeth cut by the planing process. The stationary supporting stud is made a force fit in the bores in the hubs of the housing and is secured against end-

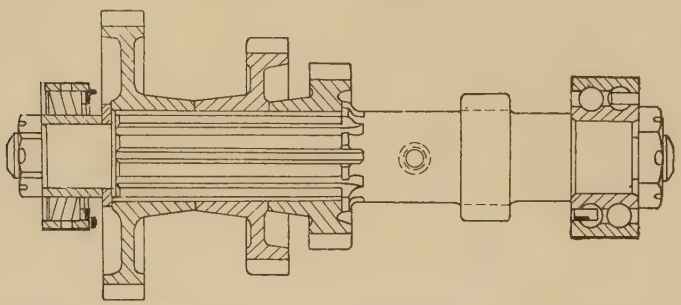


Fig. 16.—Secondary-Shaft Assembly with Gears and Bearings.

wise motion therein by means of a set screw or similar device. If the housing is of aluminum, the supporting end of the shaft near the low-speed pinion is often enlarged in diameter so as to keep down the unit pressure on the aluminum surface and prevent loosening. Where roller bearings are used this also facilitates assembling.

The cluster gear may be fitted either with bronze bushings or with flexible roller bearings. Roller bearings are generally considered preferable, on account of the reduced rate of wear and the better maintenance of proper mesh and quiet operation, and as a rule plain bushings are used only in the lower-priced constructions. In the case of a cluster gear a number of holes are drilled through the wall of the hub so that the oil in the case may readily reach the bearing surfaces. There is no appreciable end thrust on the secondary assembly, but where the case is an aluminum casting,

thrust washers are generally provided for the ends of the cluster gear to bear against.

Reverse Gear Arrangements—Various arrangements of gears for obtaining the reverse motion are in use. The most common is that already illustrated in Fig. 4, in which the secondary shaft carries a reversing pinion sufficiently smaller than the low-speed pinion to allow the low-speed gear to clear it when shifted opposite it. This reverse pinion meshes with the reverse idler on a special shaft mounted parallel with the primary and secondary shafts.

Another arrangement particularly adapted to four-speed gear boxes is illustrated in Fig. 9. There the reverse idlers are so arranged that they can be shifted axially. With the low-speed gear in the neutral position, the reverse idlers are

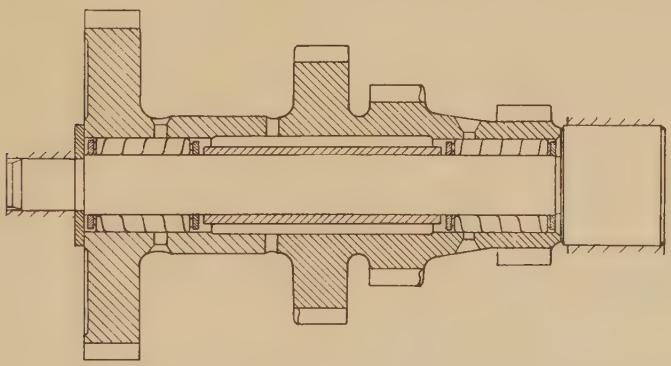


FIG. 17.—SECONDARY GEAR-CLUSTER ON STATIONARY SPINDLE.

slid forward until they are fully meshed with the low-speed gear and pinion, respectively.

A third arrangement is illustrated in Fig. 18, in which A is a pinion of double width serving for both the low gear and the reverse; B is the low-speed and reverse gear, and R_1, R_2 are reversing idlers on a special short shaft. Sliding gear B is shown in the position corresponding to the reverse motion. By sliding it to the left until it meshes with A, the low forward speed is obtained. One advantage offered by the arrangement Fig. 18 over that of Fig. 4 is that with the former there is less strain on bearing II (at the rear end of the secondary shaft) when the reverse gear is in operation.

Housings—The housings of geared transmissions are made of aluminum alloy, of cast iron, or of a special grade of high

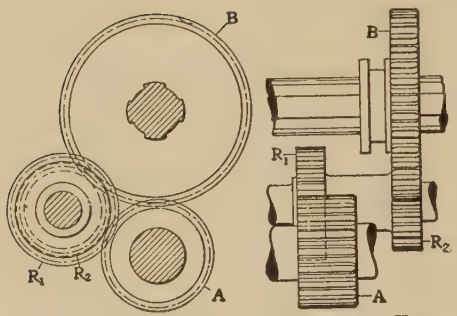


FIG. 18.—REVERSING GEAR WITH TWO IDLERS.

tensile cast iron known as semi-steel. Aluminum alloy gear-cases predominate in Europe, but in the United States they are limited to a small number of the more expensive cars, on account of the higher cost. All of the passenger cars in the medium and low-price classes have cast iron cases. The advantages of aluminum are its lower weight and its greater machinability.

The secondary shaft is now generally placed below the primary shaft, which gives a transmission housing that is comparatively deep and narrow. The reason why preference is given to this arrangement over the alternate one of having the two shafts in the same horizontal plane is that it permits of carrying a considerable amount of oil in the case without risk of leakage. The housing itself is always in a single casting with a large opening in the top through which the gears are inserted and which is closed by a cover plate. Where the transmission is to be a unit with the engine, a clutch housing must be combined with the gear housing. This is usually cast integral with it, but in a few instances it is made a separate casting bolted to the transmission housing. The dimensions of the joints between clutch housings and flywheel bell housings have been standardized by the S. A. E. (see Appendix). Another possibility consists in making the flywheel housing and clutch housing of an upper part of cast iron by which the transmission housing is supported from the crankcase, and a lower part of sheet metal.

The wall thickness is generally made 3/16 in. in the smaller (passenger car) and ¼ in. in the larger (truck and bus) transmission housings. A flange is run around the top opening to give a seat for the cover plate, of from ⅜ to ¾ in. in width, the cover being held in place by 5/16-in. or ⅜-in. cap screws spaced sufficiently closely to make the joint oil tight. If a transmission brake is to be used, the case must be provided with bosses or lugs in which the brake anchorage post and the post for the brake contracting or expanding mechanism can be supported. Housings for truck transmissions are often cast with an opening over which a power

take-off can be fitted, and dimensions for this opening have been standardized by the S. A. E. (see Appendix). In general it is advisable to design transmission housings with a minimum of flat surfaces as these tend to act as sounding boards and to intensify any noises due to the gear teeth. In transmissions for large trucks and buses the two shafts often lie in a horizontal plane.

Separately-mounted transmissions must have their housings provided with supporting lugs by which they can be hung from cross members of the frame. The bearing hubs in the housing are provided with radial ribs to give them a better support. The bottom of the housing is generally made to slope toward a point underneath the constant-mesh gear, at which point a drain plug is located. An oil filler neck is preferably cast integral with the housing on one side, at such a height that the oil will flow from it when the proper height in the case has been reached.

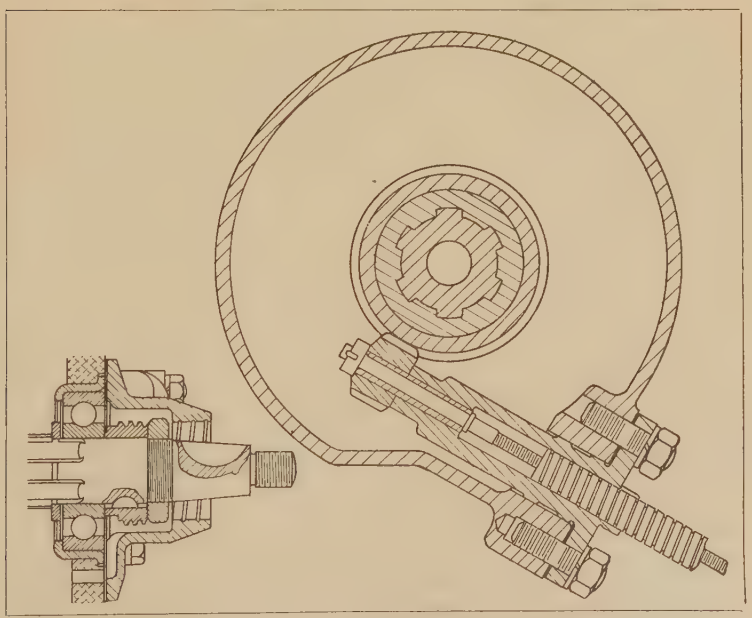


FIG. 19.—ARRANGEMENT OF SPEEDOMETER DRIVE ON TRANSMISSION DRIVE SHAFT.

(Small view shows mounting of speedometer drive pinion on shaft in another design.)

Speedometer Drive—The drive for the speedometer is now generally taken from the tailshaft of the transmission. As illustrated in Fig. 19, a helical gear is fixed to the tailshaft outside the transmission housing and a special housing for this gear is bolted to the rear of the transmission case. The shaft of the driven helical gear is supported in a bearing bolted to the wall of this housing.

Fig. 20 shows a simple method of locking the secondary shaft and reverse idler spindles against endwise motion. The spindles are made to project slightly from the end of the housing and have a slot milled in each of them. A piece of strip steel whose ends fit into the two slots is secured to the housing by a cap screw with lock washer.

Lubrication—The gears and bearings of automotive transmissions are lubricated by filling their cases with lubricant to a certain predetermined level, either transmission oil or

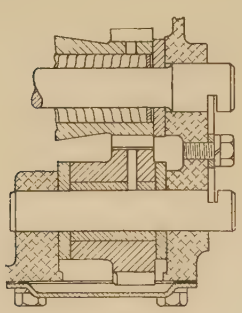


Fig. 20 (left).—Method of Securing Transmission Shafts Endwise.

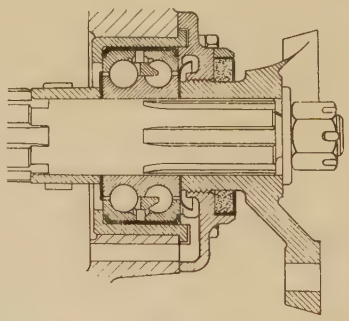


Fig. 21 (right).—Oil-Guards on Transmission Drive Shaft.

semi-fluid oil being used. Oil is introduced through an oil filler which may be formed on the cover plate but is preferably cast on the housing at one side and at such a height that when the oil overflows from its filler it is at the correct height in the case. A drain hole closed by a screw plug is provided at the lowest point of the case underneath the constant-mesh gear.

Oil Guards—With proper design there is no possibility of oil or grease working through the bearings of the secondary shaft, but the bearings of the driving shaft and splined shaft must be provided with suitable devices to keep the lubricant in. Three different types of such devices are illus-

trated in Fig. 21. First there is an oil slinger, a dished sheet metal stamping clamped between the inner race of the ball bearing and the splined fitting on the shaft. This throws off any oil accumulating on it, from its edge, whence it returns to the case through a suitable return passage. Next there is an oil scraper of the helix-type. The cap mounted over the outside of the bearing has a screw-thread turned in its bore which is a very close fit over the hub of the splined fitting. This is of such hand that with the normal direction of the splined shaft, any oil creeping over the hub of the fitting is moved by the helical scraper or thread toward the inside of the housing. Finally, there is a felt packing on the outside of the bearing cover.

It has been found that much of the trouble from oil or grease being forced through the bearings of transmissions is due to pressure in the housing, resulting from heat generated by the churning of the lubricant by the revolving gears, as well as by tooth friction, and to relieve such pressure, transmission cases are sometimes provided with breathers, the same as engine crankcases.

Sliding Mechanism—The mechanism by means of which the various gears are slid into and out of mesh comprises a lever, a number of slider bars having bearings in the wall of the case or cover, and a fork or yoke for each slider bar, fastened to the bar and engaging into the groove on the sliding gear. The lower end of the shift lever is capable of sliding motion in one plane and of rocking motion in planes at right angles thereto, or else of a universal angular motion around a fixed fulcrum point. The former arrangement is the older one, but it has now been practically displaced by the latter. A typical slider mechanism for a three-speed transmission is shown in Fig. 22.

Two slider bars are required for transmissions with three forward speeds and one reverse, while three bars are required for four-speed transmissions. These bars are arranged close together in a horizontal plane. Notches are cut either in the bars themselves or in the hubs of the shifter forks on them, into which the lower end of the shifter lever is adapted to engage, this engagement being effected by a sideward motion of the top of the lever. After the engagement has been effected, forward motion of the lever will engage one gear speed, and rearward motion another. Thus, in the case of the ordinary three-speed transmission, the direct drive high and the intermediate speed are obtained through one slider bar, and the low forward and the reverse speed are obtained through the other. Some means must be provided to prevent

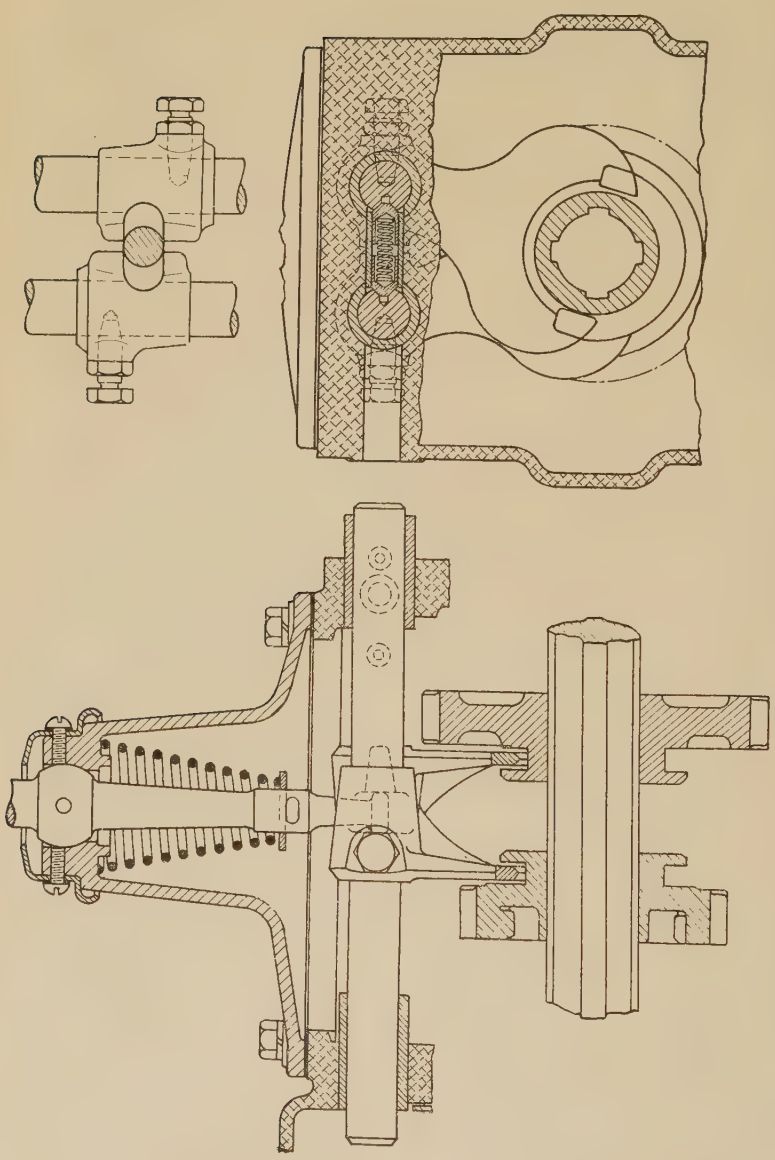


FIG. 22.—SHIFTER MECHANISM FOR TRANSMISSION WITH THREE FORWARD SPEEDS AND REVERSE (HUPP).

the slider bar not connected with the shift lever, and the gear connected thereto, from drifting while the gear is in operation, as an accidental engagement of two pairs of gears at the same time would wreck the transmission. To this end the slider bars are provided each with three notches, corresponding to the two positions of engagement and the neutral position in between, into which the nose of a spring-pressed dog may engage. This dog serves both as a position finder and as a lock. In some cases a positive latch is provided. When the shift lever is moved sideways to pick up one of the slider bars it moves a latch bolt which positively locks the other bar against endwise motion.

Lock-out Latch—With four-speed gears having three slider bars there is some danger that when the operator slides

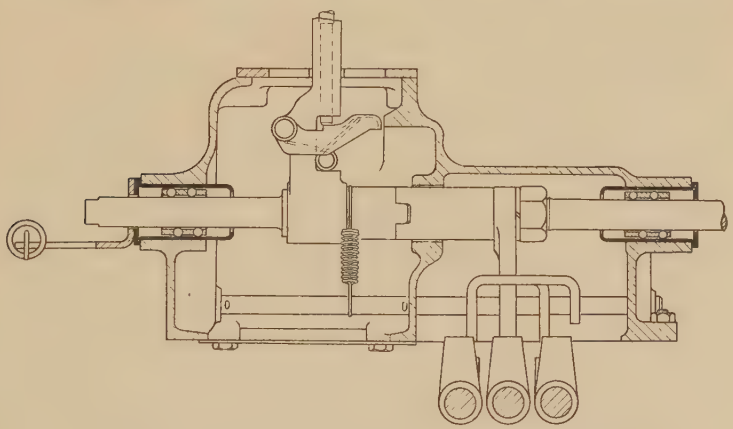


FIG. 23.—REVERSE-GEAR LOCK-OUT LATCH.

the shift lever sideways to engage the slider bar controlling the two lower forward speeds, he will inadvertently slide it too far and pick up the bar controlling the reverse gears. This might under certain conditions result in a collision, especially in crowded traffic, and to prevent it, some cars have a lock-out latch on the shift mechanism, as illustrated in Fig. 23. With this it is necessary, before the shift lever can be slid over sideways into the position where the reverse slider bar is engaged, to press on the latch rod projecting up through the center of the shift lever handle. This depresses the latch and permits the lever to engage the reverse slider bar. Such a lock-out latch is used also with four-speed gears

of the internal gear type in which the low-speed gear is used for emergencies only and is therefore latched out.

Shift Lever Mounting—The opening in the gearcase through which the gears are introduced is closed by a cover plate, which in most cases carries both the shift lever mounting and the bearings for the slider rods. A typical cover with shift lever mounting is shown in Fig. 24. The lever is of what is known as the cane type and has a spherical mounting at the top of a column cast integral with the cover. The two halves of the spherical bearing are pressed together by a spring, and between them there is a washer, clamped between the cap-shaped upper half of the bearing and the top of the column, with a key on its inner edge which

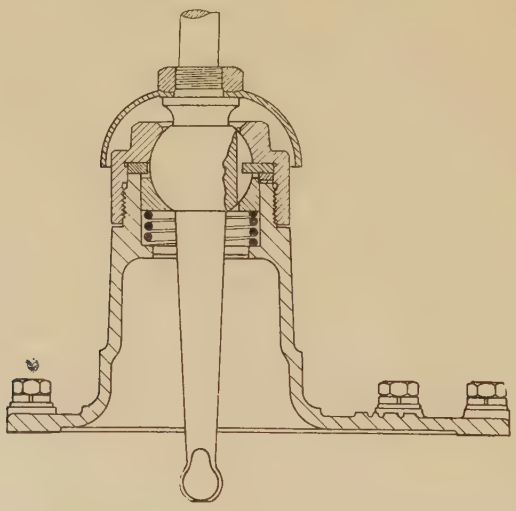


FIG. 24.—SHIFT-LEVER MOUNTING.

enters a milled slot in the ball of the lever, to prevent the latter from rotating around its axis.

Indirect Fourth Speed—A number of four-speed transmissions have been used on large passenger cars in the past in which the third speed was the direct drive and the fourth speed an over-drive, that is, higher than the direct drive. The only difference between a normal four-speed transmission and one with an over-geared fourth is that in the latter, one pair of meshing gears produces a speed multiplication from the secondary to the splined shaft which is greater than

the inverse of the speed reduction through the constant-mesh gears, whereas in the former, if there is a speed multiplication from the secondary to the splined shaft, it is much less than the inverse of the speed reduction of the constant-mesh gears. The theory on which the design of this transmission was based is that the direct drive, which is the most efficient, the most silent and generally the most satisfactory from the occupants' point of view, should be used as much as possible, and it can be used more if the rear axle ratio is made smaller, which is possible with this type of transmission. With a smaller rear axle ratio the car will have greater acceleration on direct drive, and it also can take hills on the direct drive that would otherwise call for a shift to a lower gear. These are the advantages of the indirect fourth gear or over-drive. The disadvantages are that the car is driven most of the time in third speed, and with the smaller rear axle ratio the engine load factor is lower, which means less efficient operation of the engine (lower fuel economy) and high wear and tear due to high engine speeds. Moreover, the pitchline velocity of the gears for the fourth speed is very high, and the gears generally operate with considerable noise. The disadvantages evidently overbalanced the advantages, as the gear was abandoned.

Transmissions with Internal Gears—In 1926 a type of transmission first made its appearance in which two pairs of internal gears are used to produce a "geared speed" that rivals the direct drive in point of silence of operation, and is almost equal to it in respect to transmission efficiency. This new transmission was put forward in the first place as a four-speed type to take the place of the three-speed transmission previously used on practically all American passenger cars. The demand of American motorists for constantly greater acceleration on high gear, and for the ability to take grades of 10 per cent and more on direct drive, had led to the use of large rear axle ratios in combination with engines of large displacement, which resulted in rough operation of engines at higher car speeds, reduced engine life and increased fuel consumption. The four-speed transmission with two silent ranges—obtained through the use of internal gears—made possible a smaller rear axle reduction, which obviated the disadvantages alluded to; and an easy shift from third to fourth and vice versa, due to a comparatively small difference between these two speeds and the use of positive clutches for engaging both the third and the fourth speeds. The only disadvantages were the somewhat more frequent shifts and the increased cost of the four-speed

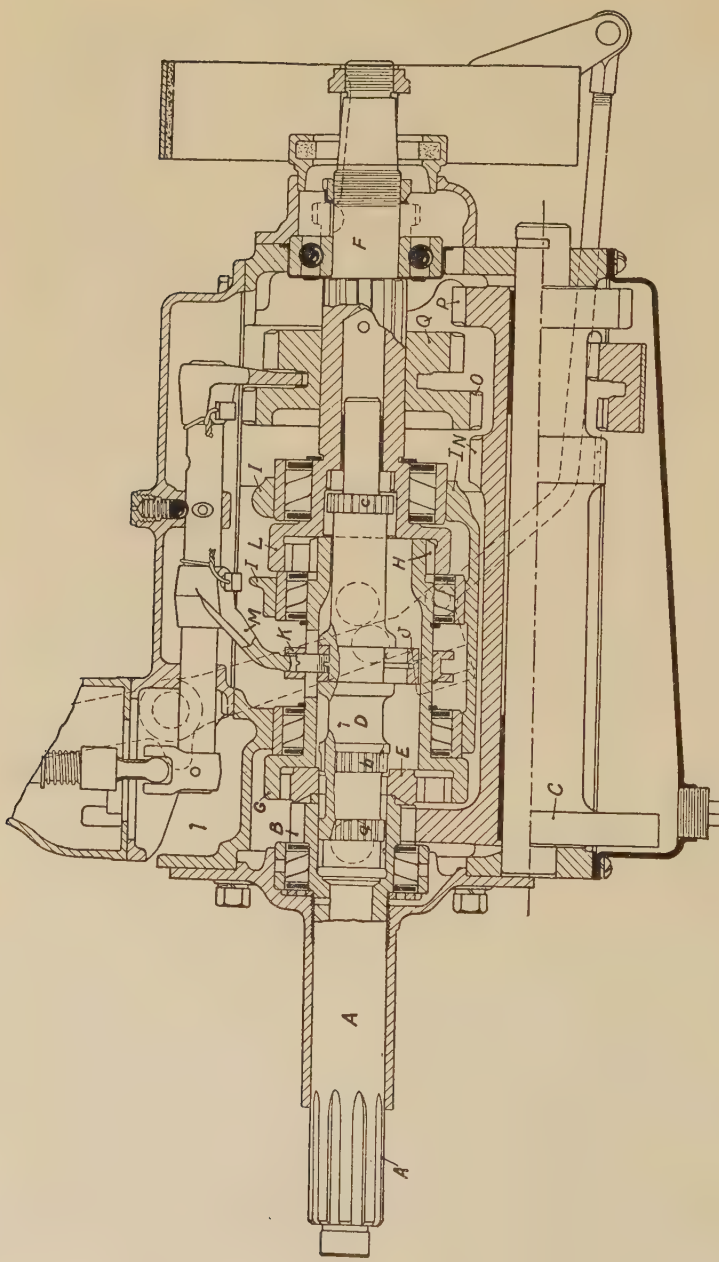


Fig. 25.—Four-Speed Transmission with Internal Gears (Warner Hi-Flex).

gear. Later on three-speed gears with an internal gear unit were also brought out.

Warner Hi-Flex Gear—Several different makes of gears of this general type are being manufactured, the differences pertaining chiefly to the method of supporting the internal gear unit or spool gear, and to the method of operating the clutch for the direct drive and the internal-geared drive. The general arrangement of such a transmission may be explained by reference to the Warner Hi-Flex, illustrated in Fig. 25.

The clutch shaft A at its rear end carries the constant-mesh pinion B, which latter meshes with the constant-mesh gear C of the secondary cluster gear which revolves on plain bearings on a spindle fixed in the gear housing. Clutch shaft A is drilled out from the rear end and internally splined, and within the hole in it is located the forward end of a floating shaft D which is splined at both ends and also some distance back of the forward end. By means of the forward splines *a* it remains in driving connection with the clutch shaft at all times, although it can be shifted axially a certain distance by means of mechanism to be described. The second set of splines *b* serves to place the floating shaft D in driving connection with the third speed drive pinion E, while the set of splines *c* at the rear end of the shaft connects it with the tail shaft F when the floating shaft is moved toward the rear. In the drawing the floating shaft is shown in the central or neutral position, in which it is connected neither to the third-speed drive pinion nor to the tail shaft.

The floating shaft is surrounded by a shell-like member comprising the internal gear G and the spur pinion H and referred to as a double internal gear. This member is supported in two Hyatt bearings mounted in a bearing support I forming a bridge across the main housing. At about the middle of its length the floating shaft is enlarged and has a groove for a shifter collar cut in it. Into the shifter collar groove in shaft D fits eccentrically the collar J which connects by pins extending radially through longitudinal slots in the wall of the hollow member G-H with a grooved collar K on the outside of this member. Into the groove of this collar engages a fork M mounted on one of the shifter bars.

When the floating shaft D is shifted forward, the third-speed drive pinion E is secured to it by means of splines b, thus giving the third speed, while if shaft D is shifted toward the rear, the splines *c* connect it with tail shaft F, and since the floating shaft is at all times in driving relation with clutch shaft A through the splines *a*, the tail shaft is then connected to the clutch shaft through the floating shaft, and the drive is direct.

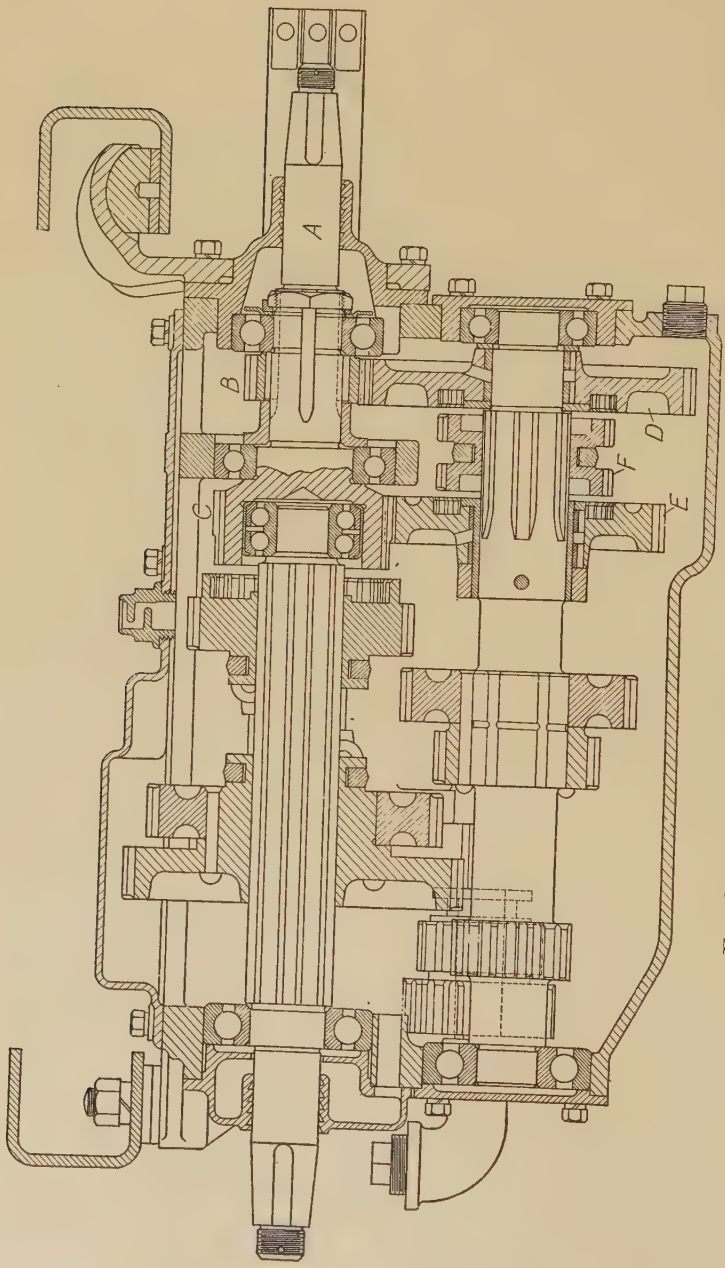


FIG. 26.—G. M. T. MULTI-SPEED TRUCK TRANSMISSION.

It will be noticed that the tail shaft F at its forward end carries the internal gear L, with which meshes the pinion H. Thus for the third forward speed the drive is from the clutch shaft through the internal combination EG to the double internal and thence through the internal gear combination HL to the tail shaft F. Except when driving on third the internal train runs idly.

First and second speed and reverse are obtained in the usual manner, except that the secondary pinion for the second speed is located close to the rear wall of the housing and the pinion for the first speed and reverse further forward, which is the reverse of the conventional arrangement. For the first speed the power is transmitted from B to C and from N to O, while for second speed it is transmitted from B to C and from P to Q.

Gear Problems—There are a number of special problems involved in the design of a set of internal gears for a transmission of the type described. Since the total reduction of the two pairs of gears must not be greater than 1.40 to 1.42, the ratio of each cannot be greater than 1.18 or 1.19. The diameters of the internal gears are limited by considerations of bulk and weight, as well as of pitchline velocity. This leads to the adoption of fine-pitch gears (generally 10–12 pitch) with large pressure angles ($27\frac{1}{2}°$). It is desirable to keep the backlash between teeth of the internal gears very small, as this relieves shocks and facilitates shifting. In order that the internal gears may operate silently, they must be flooded with oil, and an oil circulating system is generally provided. The oil thrown off by one of the gears in the rear part of the housing can be caught in a trough on the wall of the housing near the top and conducted by suitable passages into the compartment in which the internal gears are located, in which a constant level of lubricant is maintained by means of an overflow dam.

Multiple Speed Transmissions—It has been pointed out already that for heavy trucks used for a wide variety of work, transmissions with as many as seven forward speeds are sometimes used. This number of speeds can be readily obtained without material increase in bulk and weight by providing the ordinary four-speed transmission with an extra pair of first-reduction gears, so that the secondary shaft can be run at two different speeds relative to the transmission drive shaft. If both of these pairs of gears remain constantly in mesh, the driven members must be loose on the secondary shaft, and positive clutches must be provided by means of which one or the other can be placed in driving connection

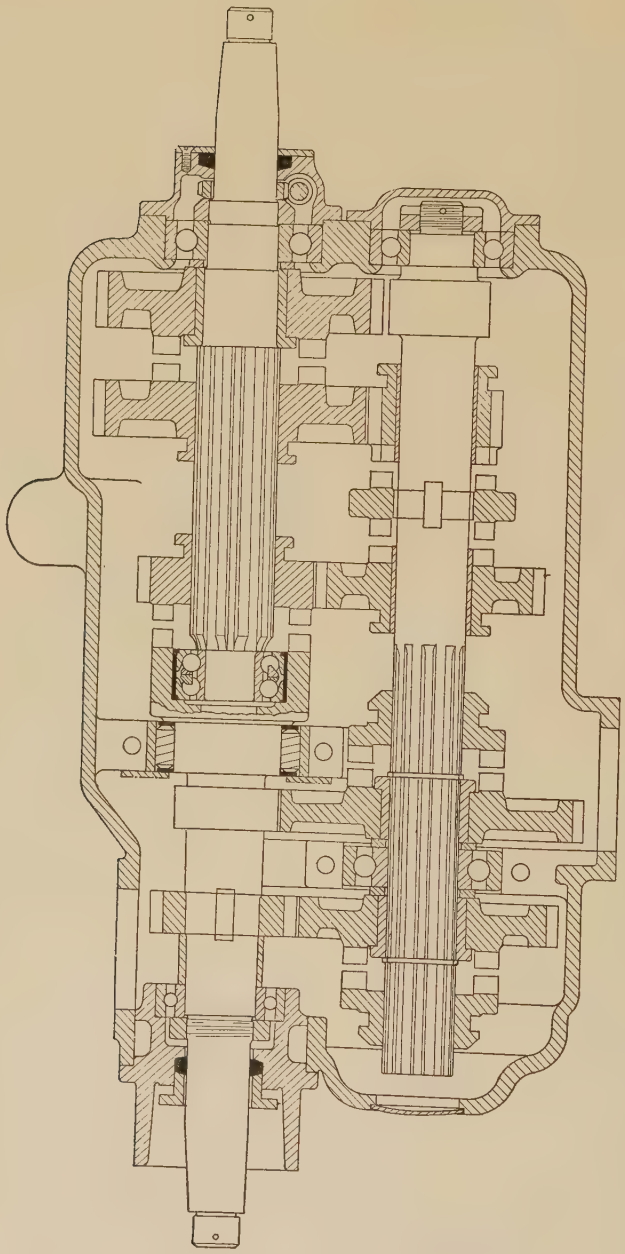


FIG. 27.—COTTA CONSTANT-MESH TRUCK TRANSMISSION.

therewith at the will of the operator. The two pairs of gears can also be engaged and disengaged by sliding them into and out of mesh, in which case they are no longer constant-mesh, but first-reduction gears.

A transmission of this type, due to the General Motors Truck Company, is illustrated in Fig. 26. The transmission drive shaft A is mounted in two ball bearings and one of the constant-mesh pinions, B, is located between them, while the other, C, overhangs the bearings. The two constant-mesh gears, D and E, are free on the secondary shaft but can be fastened thereto by the sliding clutch F. In this transmission there are four sliding units, viz., the clutch member just referred to, the direct-drive clutch and third-speed gear, the low- and second-speed gears, and the reverse idler. There is only one direct drive, but each pair of gears meshed by sliding can be used together with either of the pairs of "first-reduction" gears, thus giving six forward geared speeds and one direct drive, and also two reverse speeds.

Heavy Duty Transmissions—In transmissions for heavy-duty work, like those used on farm tractors, railcars and gasoline locomotives, where practically the full engine torque may have to be transmitted for long periods at a time, the gears must be made of more liberal size, depending upon the material from which they are made and the relative frequency of use of the particular pair. If made of case-hardened or oil-hardened steel, the limiting compression stress in the teeth, as calculated by the method previously explained, should not exceed 100,000 lbs. per sq. in. for gears that may be used continuously, nor 150,000 lbs. per sq. in. for gears used less frequently. Similarly, if the gears are made of forged steel, unhardened, the limiting compression stress should not exceed 50,000 lbs. per sq. in. for the pairs most frequently used, nor 75,000 lbs. per sq. in. for the pairs less frequently used. These figures apply to first-reduction gears, with pitch-line velocities of from 1000–1500 ft. p. m. For final-reduction gears for tractors, the pitchline velocities of which are much lower, the compressive stresses under full engine load through the largest gear reduction (the reserve gear) may be carried as high as 200,000 lbs. p. sq. in. with hardened, and 100,000 lbs. p. sq. in. with unhardened gears.

Constant-Mesh Type Transmission—Fig. 27 shows a constant-mesh transmission designed by Charles Cotta. It is a truck transmission and gives five forward speeds and two reverse speeds, there being two pair of first-reduction gears. Of each pair of gears one is splined to its shaft while the other is free to revolve but may be locked to its shaft by

FIGS. 28 AND 29.—OVER-RUNNING CLUTCH FOR MOTOR-VEHICLE DRIVE.

means of a positive clutch. In some cases the clutch member is shifted toward the gear, while in others the gear is shifted toward the clutch member, the mating gear then being shifted simultaneously. Whereas most transmissions with more than four forward speeds require two levers for operating them, the entire control of this transmission is by a single lever.

Efficiency of Sliding Gear Transmissions—The efficiency with which a sliding gear combination transmits power is quite high. It is not constant but varies with the torque and the speed. Some of the losses, such as that due to the churning of the lubricant in the case, are constant, for which reason the efficiency is lower when only a small amount of power is being transmitted. Published data of transmission efficiency tests indicate that on the direct drive the efficiency reaches about 95 per cent when one quarter the full load torque is being transmitted and varies between 95 and 98 per cent over the remainder of the torque range; on second speed the efficiency attains about 90 per cent at one-quarter full torque and varies between 90 and 95 per cent over the remainder of the torque range; in low speed the efficiency reaches about 85 per cent at one-quarter full torque and varies between 85 and 94 per cent over the remainder of the torque range. With plain bearings on the secondary shaft the efficiencies may be expected to be somewhat lower. The viscosity of the lubricant in the transmission case also has a bearing on the efficiency.

Over-running Clutches—In Europe, and particularly in England, some of the smaller cars in recent years have been fitted with over-running clutches behind the transmission. With such a device the car will coast automatically when the engine is throttled down. Of course, a car can be made to coast by merely holding out the clutch (and throttling the engine), but this compels the operator to press down on the clutch pedal continuously, which is tiring; and, besides, the clutch throw-out bearings on many cars soon give trouble if the clutch is held out for long periods.

Automatic coasting or free-wheeling is made possible also by the provision of an automatic friction clutch—in place of the conventional one—and such a clutch was brought out by the Nationale Automobil-Gesellschaft in Germany in 1928. A spring normally holds the clutch out of engagement, but when the engine is speeded up, centrifugal weights which rotate with the flywheel, overcome the pressure of the springs and engage the clutch.

The use of an over-running clutch back of the transmission has the advantage that it facilitates gear-shifting. When

the gears are being shifted, that part of the transmission back of the point where the shift occurs has very little inertia, and can be slowed down quickly, as usually required in shifting to a lower gear; hence, shifting can be effected even without unclutching, by merely throttling down the engine. Another advantage, which is really incidental to the one already mentioned, that the over-running clutch permits of coasting in comfort, is that it tends toward economy in fuel consumption.

With such an over-running clutch it would be impossible to utilize the friction of the engine for slowing down the car. This characteristic of the conventional car is quite desirable, as it tends to lessen wear of the brake lining, and to retain it where an over-running clutch is fitted it is usual to provide the clutch with a device by means of which it can be locked from the seat at the will of the operator.

A typical design of over-running clutch, as fitted to several English cars in 1928, is shown in two sectional views in Figs. 28 and 29. It is contained in a housing which forms a rearward extension of the transmission housing, and consists of the over-running roller clutch A, the driving member of which is attached to the rear end of an extension of the main gear shaft, while the driven member is connected to a flange for the front universal of the propeller shaft.

When desired—to secure braking effect from the throttled engine, for instance—the over-running clutch can be locked by means of the control lever shown, the latter being located near the gearshift lever. A fork on the lever shaft engages a grooved flange having dog teeth B that can be meshed with corresponding teeth on the front face of the driven member of the clutch; to facilitate the engagement of the dog teeth a spring-backed friction plate C is provided to equalize the speeds of the two members before the dog teeth meet.

CHAPTER V

Propeller Shafts and Universal Joints

The transmission of power from the change-speed gear to the rear axle or the rear wheels involves some difficulty, because of the relative change in position of the two with spring action. There are essentially two methods of transmitting power from a spring-supported gearbox to the rear axle or rear wheels, namely, by chains and by a propeller shaft with universal joints (usually called Cardan shaft in Europe). The chain came into use first, but it has now been almost entirely superseded by the propeller shaft. There is little doubt that when the chain is clean, well oiled and well adjusted, it is the most efficient means of transmission yet devised, but difficulty arises in keeping it so. In spite of numerous attempts, the problem of enclosing the chain in a dust- and oil-proof case was never satisfactorily solved, hence it must be run exposed. Since it is covered with grease when first applied, it soon collects dust and grit, which gets into the pin bearings and causes rather rapid wear. This results in "stretching" of the chain and in whipping of same when the vehicle is being driven at certain speeds. It then becomes necessary to make adjustments. In spite of these defects, the chain drive gives fairly satisfactory service when properly cared for, but the great majority of designers prefer the shaft drive in which all wearing surfaces are protected by dust-proof housings, so they can be properly lubricated.

Shaft Materials—Propeller shafts can be made of the same materials as rear axle shafts, though it is not so essential to use high-grade alloy steel for the former as for the latter. The materials used for these shafts include carbon steel, nickel steel, chrome-nickel steel, chrome-vanadium steel and chrome-molybdenum steel, the carbon content in all of these steels usually varying between 0.40 and 0.50 per cent. The physical property of these steels which determines their suitability for this purpose is their resistance to shear in torsion as determined in a torsion testing machine. Unfor-

tunately the data available on this quality of the steels usually employed for this purpose are very meagre.

By means of a torsion testing machine data can be determined which permits of plotting a torsional stress-strain curve for the particular specimen. This curve shows the elastic limit and the ultimate strength of the specimen as a whole, and the unit elastic limit to shear in torsion can then be determined by means of the equation

$$M = 0.196\, d^3 S$$

where M is the torsion moment in lb.-in.; d, the diameter of the specimen in in. and S, the shearing stress. When a cylindrical specimen is subjected to torsion, the stress on the fibers varies in direct proportion to their distance from the axis, being zero at the axis and a maximum at the surface. This direct relationship between distance from axis and stress holds only as long as the stress nowhere exceeds the elastic limit. Therefore, if we know the moment required to stress a certain specimen to the elastic limit, by using the above equation we can calculate the elastic limit of the material under torsional shear. The same equation cannot properly be used for determining the ultimate strength of the material from the breaking moment. We can, however, use it to determine what we may call the apparent ultimate strength, which is useful in comparing different materials. The actual ultimate strength under torsional shear is lower than the value found by the application of the above equation.

Torsional Strength of Materials—Following are data on the elastic limit and apparent breaking strength under torsional shear of different steels, which the author has compiled from different sources.

In a paper presented to the Institution of Automobile Engineers by C. E. Larard in 1911, torsional strength data are given for a mild steel containing C, 0.12 per cent; S, 0.034 per cent; P, 0.031 per cent; Mn, 0.41 per cent and Si, 0.006 per cent. This steel is not suited for propeller shafts, but splined shafts of it were used, after case-hardening, for transmissions. The average elastic limit of shafts of from 1 to 2 in. diameter was close to 10,000 lbs. per sq. in., and the apparent breaking strength, close to 60,000 lbs. per sq. in. The enormous difference between the elastic limit and the ultimate strength will be noted. This is due to the fact that the apparent breaking strength is a fictitious value and the real breaking strength is lower. It was pointed out in the discussion that the elastic limit would have been found materially higher if the specimens had been case-hardened before

the torsion tests were made, and this prediction has since been verified by John Miller, metallurgist of the Pierce Arrow Motor Car Company. Since the stress in a round specimen under torsion increases from the axis outward, failure must begin at the surface if the material is uniform throughout. Therefore, by improving the mechanical properties of the case or surface layer the strength of the whole shaft can be materially increased.

Mr. Larard also gave test figures for splined shafts of a steel containing about 3 per cent of nickel. The carbon content is not given, but presumably it was about 0.40 per cent. In calculating the elastic limit of the material from the test data the author has made use of relations found by other investigators between the strengths of splined shafts and of plain shafts of diameters equal to the outside and bottom diameters of the splined shafts, respectively. In the normal state (or as received from the forge) this steel showed an average elastic limit under torsion of 28,000 lbs. per sq. in. When the steel was heat treated by being heated to 1550° F. and quenched in oil, the average elastic limit in torsion increased to 55,000 lbs. per sq. in. In all cases the smaller shafts (1 in.) showed slightly greater values than the larger ones (2 in.). The apparent ultimate strengths of the nickel steel tested by Larard was 78,000 lbs. per sq. in. untreated and 91,500 lbs. when heat-treated as described.

Medium-Carbon Steel—C. W. Spicer of the Spicer Manufacturing Corporation presented to the S. A. E. at its 1927 summer meeting, data of some torsion test results on heat-treated carbon steels, the tests having been made on both splined and plain specimens. Some of the plain specimens had a diameter equal to the bottom diameter of the splined sections, while others had a diameter equal to the top diameter of same. It was found that the elastic limit of the splined specimens was even less than that of a plain round specimen of a diameter equal to the bottom diameter of the splined section, but that the ultimate strength of the splined section was much greater than that of the small plain section, and only about 10 per cent less than that of the full round section equal in diameter to the top diameter of the splined section. The explanation of this phenomenon, which may seem puzzling at first, is that, since the splines add very little to the resistance to torsion, the splined specimen will be twisted nearly as much as the small solid specimen by a given torque, but the stress at the top of the splines, which is further from the axis, will be greater than the stress at the surface of the solid specimen,

and the elastic limit of the specimen is determined by the stress on the most highly stressed fiber. When the ultimate strength is approached the splines are so distorted that their material is largely under tension, and this changes conditions.

The material employed in these tests by Mr. Spicer was S. A. E. No. 1045 steel (0.44 C, 0.630 Mn, 0.034 S, 0.017 P). It was heated to 1625° F. and cooled in the air; reheated to 1525° F. and quenched in water; reheated and drawn at 1200° F. for 30 mins. The results showed an average elastic limit in torsion of 45,000 lbs. per sq. in. and an average apparent ultimate strength of 96,000 lbs. per sq. in.

Chrome-Nickel Steel—Some of the Chrysler models in 1928 had rear axle shafts made of a steel very similar to S. A. E. No. 3145, except that the carbon range was held between 0.43 and 0.48 instead of between 0.40 and 0.50. These shafts were manufactured by the rolling process. The heat treatment consisted in annealing at 1550°–1600° F., oil-quenching from 1550°–1600° F. and drawing at approximately 1000° F. to obtain the specified hardness (286–321 Brinell for one model and 300–340 for another). When drawn to give a Brinell hardness of 321, the steel showed an elastic limit under tension of 132,300 lbs. per sq. in. and an ultimate strength of 150,000 lbs. per sq. in. Torsion tests on a number of axles with an average Brinell hardness of 325 showed an average elastic limit of 94,400 lbs. per sq. in. and an average apparent ultimate strength of 134,000 lbs. per sq. in. The average angle of torsion at the elastic limit was 35° and at failure, 444°.

In the more powerful Chrysler models steel No. 3435 was used for axle shafts—a high-alloy chrome-nickel steel. Shafts made from this steel were annealed and rough-machined; they were oil-quenched from 1500°–1550° F. and drawn at 850°–900° F. to give the specified hardness (340–375 Brinell for one model and somewhat greater for another). Only the cutting of the splines at one end and of the keyway at the other was deferred until after heat treatment. This steel showed 140,000 lbs. per sq. in. elastic limit and 184,200 lbs. per sq. in. ultimate strength under tension when drawn to 387 Brinell. Torsion tests on shafts drawn to an average of 387 Brinell showed an average elastic limit of 127,200 lbs. per sq. in. and an average apparent ultimate strength of 144,000 lbs. per sq. in. The angle of torsion of the shafts averaged 42° at the elastic limit and 150° at the break.

Chrome-Molybdenum Steel—Chrome-molybdenum steel of the following composition was used by the Hupp Motor Car Co. in 1928: 0.35–0.45 C, 0.40–0.70 Mn, 0.045 max. Ph,

0.045 max. S, 0.30–0.40 Mb, 0.80–1.10 Cr. This steel was normalized at 1625°–1650° F., quenched in oil from 1500°–1550° F., heated to 1025°–1075° F. and cooled in air. The Brinell range specified was 285–321. When thus heat treated the steel showed an elastic limit in tension of 140,000 lbs. per sq. in. and an ultimate strength of 175,000 lbs. per sq. in. Torsion tests on a number of axles showed an average elastic limit of 82,300 lbs. per sq. in. and an average apparent ultimate strength of 127,000 lbs. per sq. in.

In the design of propeller shafts a factor of safety of 3 to 4, based on the ultimate strength, should be figured with in determining the necessary diameter of the weakest section, under a torque equal to the maximum engine torque multiplied by the low gear ratio.

Basis for Factor of Safety—This brings up the question as to whether the factor of safety should be based on the elastic limit or the breaking strength. Forty years ago it was the universal custom to base strength calculations on the ultimate strength of the material. Later it was argued that, since any part in which the elastic limit of the material is periodically exceeded must ultimately fail, the maximum safe stress depends really on the elastic limit, and a practice of basing strength calculations on this value grew up.

During the past decade there have been very extensive researches on the fatigue of metals and on the so-called endurance limit of various metals and alloys. This is the maximum repetition stress which a metal will withstand indefinitely without failure, and is determined by testing specimens of the material in endurance machines under various higher stresses, and then plotting stresses against numbers of repetitions before failure. The stress at which the curve becomes horizontal or parallel to the number-of-repetitions axis is the endurance limit. It has been found that the endurance limit, which is evidently the maximum safe stress, is more nearly proportional to the ultimate strength and the hardness than to the elastic limit of the material, hence, for parts subjected to repetition stresses at least, the ultimate strength seems to be the most logical basis for strength calculations in design.

Critical Speeds—Under certain special conditions propeller shafts are subjected to supplementary stresses which must be guarded against. Not long after the shaft drive had come into common use, trouble began to develop from severe vibration, resulting in permanent bending and even breaking of propeller shafts, at certain critical speeds, especially on cars with unit powerplants or transmission axles, which necessitate the use of exceptionally long shafts. The phenomenon

was not entirely new, but had previously been observed in connection with steam turbines, which run at very much higher speeds, and a mathematical explanation had been worked out. This explanation is briefly as follows:

In spite of careful workmanship the center of mass of a revolving shaft never coincides exactly with the center of revolution. Owing to the eccentricity of the center of mass, an unbalanced, centrifugal force is produced which causes the shaft to vibrate. The phenomenon would be still more pronounced if the shaft carried a heavy disc at the middle of its length, with its center of mass outside of the axis of rotation. (See Fig. 1.) Let the center of mass be at a distance d from the axis of revolution of the shaft. Under the influence of centrifugal force the shaft at the middle of its length will deflect the distance y from its neutral position. When thus deflected there will also be an unbalanced centrifugal force acting on the mass of the shaft, which will add to the

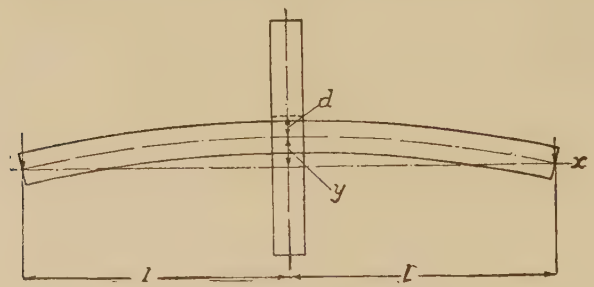


FIG. 1.—SHAFT CARRYING CENTRAL UNBALANCED DISC.

centrifugal force acting on the mass of the disc, but for the present purpose it is permissible to neglect the former. Denoting the mass of the disc by m, the centrifugal force is expressed by

$$F = m\,(y + d)\,\omega^2,$$

ω being the angular speed in radians per second.

This force is balanced by the elastic force of the shaft which is proportional to the deflection and may, therefore, be represented by

$$F_e = ay.$$

hence

$$m\,(y + e)\omega^2 = ay$$

from which it follows that

$$my\omega^2 + me\omega^2 = ay$$

and

$$y = \frac{me\omega^2}{a - m\omega^2}.$$

When $a = m\omega^2$ the deflection becomes infinite; that is, unless the vibration of the shaft is limited by bearings or guards, the shaft will break. This, therefore, is the condition defining the critical speed. There is one other exception in addition to that noted above which would preclude breaking of the shaft, and that is that the speed of revolution of the shaft varies so rapidly that it remains near the critical speed an insufficient length of time to permit of a dangerous vibration being attained. Since the equation defining the critical speed is

$$a = m\omega^2$$

the value of the critical speed is evidently

$$\omega = \sqrt{\frac{a}{m}}.$$

In order that the centrifugal force may be expressed in pounds (the angular speed ω being given in radians per second) the linear dimensions must be given in terms of the foot, and a then is the force necessary to deflect the shaft one foot at the middle of its length.

Analysis of Critical Speeds—Now consider a section dx of a freely supported shaft carrying only its own weight (Fig. 2). When the shaft rotates the shaft section dx is under the influence of two external forces, the force of gravity and centrifugal force. With the shaft proportions found in practice the former has no appreciable bending effect and may be neglected. The centrifugal force puts a load on the shaft which revolves with it and subjects it to shear and bending stresses. These latter can be determined by means of the theory of beams, the shaft being equivalent to a simple beam supported at both ends. When the shaft is in equilibrium the external (centrifugal) and internal (elastic) forces must neutralize each other in every plane.

Let the shear at the two sides of the infinitesimal section dx of the shaft be denoted by S and S' respectively. The centrifugal force on the section dx is proportional to the length dx and may be represented by $p\ dx$, where p is the centrifugal force per unit length. Then, since there must be equilibrium in the vertical plane

$$S' - S + p\ dx = 0.$$

But

$$S' - S = dS,$$

hence

$$\frac{dS}{dx} = - p.$$

Also, taking moments around the center of gravity of the section dx,

$$M' - M\,S' \frac{dx}{2} - S \frac{dx}{2} = 0,$$

and since

$$M' - M = dM$$

we have

$$dM = \frac{S' + S}{2}\, dx = S\, dx.$$

This relation can now be combined with the equation of the elastic curve of a beam, viz.,

$$\frac{d^2y}{dx^2} = \frac{- M}{EI},$$

the minus sign being used here to correspond with the designations in the cut.

Since

$$\frac{dM}{dx} = S$$

$$\frac{dS}{dx} = - p = \frac{d^2M}{dx^2} = - EI \frac{d^4y}{dx^4}.$$

Hence

$$p = EI \frac{d^4y}{dx^4} = m\omega^2(y + d).$$

The general integration of this equation gives

$$y = ae^{cx} + a'e^{-cx} + b \cos cx + b' \sin cx - d,$$

in which

$$c = + \sqrt[4]{\frac{m\omega^2}{EI}}$$

and e is the base of the natural system of logarithms. This equation covers all possible conditions of rotating shafts, and the values of the constants a a' b b' depend upon the conditions of any particular case—whether the shaft is freely supported or rigidly held in bearings, supported at both ends or at one end only, etc.

If now we take a freely supported shaft like a propeller shaft with universal joints at both ends, and if we measure the abscissas from the middle of length of the shaft, then y must be an even function of x, in order that the same value for y may be obtained for equal positive and negative values of x. The third term of the above equation for y contains a cosine and the value of the cosine is the same for a positive and negative angle of equal magnitude. The sines of positive and negative angles are alike but opposite in sign. A change in the sign of x would not give the same value of opposite signs for each of the first two terms. Consequently variations in the first two terms due to a change in the sign of x could not be compensated for by a corresponding variation in the fourth term. The conclusion to be drawn is that when

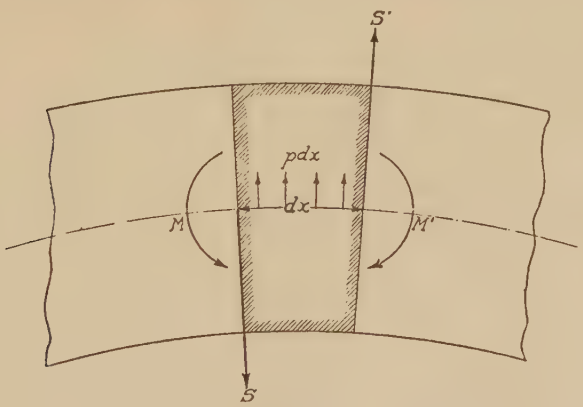


FIG. 2.—DIAGRAM OF CENTRIFUGAL AND ELASTIC FORCES.

x changes sign there is no variation in the sum of the first two terms, and no variation in the value of the fourth term. From this it follows that

$$a = a' \text{ and } b' = 0.$$

This gives us

$$y = a(e^{cx} + e^{-cx}) + b \cos cx - d.$$

Now when $x = l$ or $-l$

$$\frac{d^2y}{dx^2} = \frac{-M}{EI} = 0,$$

because $M = 0$.

Under these conditions

$$\frac{d^2y}{dx^2} = a\,(e^{cl} + e^{-cl}) - b\cos cl = 0.$$

Hence

$$a\,(e^{cl} + e^{-cl}) = b\cos cl.$$

Substituting in the above equation for y and remembering that when $x = l$, $y = 0$,

$$2a\,(e^{cl} + e^{-cl}) = d$$

$$a = \frac{d}{2\,(e^{cl} + e^{-cl})}.$$

Also

$$2b\cos cl = d$$

$$b = \frac{d}{2\cos cl}.$$

When cos cl is zero the value of b, and consequently the value of y, the deflection, becomes infinite, and the shaft runs at the critical speed. This is the case when $cl = \pi/2$, $3\pi/2$, $5\pi/2$, etc. There are, therefore, a number of critical speeds. Now, inserting the value of c in the equation for b and equating the latter to the smallest angle corresponding to a critical speed we get

$$l\sqrt[4]{\frac{m\omega^2}{EI}} = \frac{\pi}{2};$$

$$\frac{l^4 m\omega^2}{EI} = \frac{\pi^4}{16};$$

$$\omega^2 = \frac{\pi^4 EI}{16m\,l^4}.$$

in which

ω = the angular speed in radians per second,
E = the moment of inertia of the shaft section,
I = the modulus of elasticity,
m = the mass of the shaft per inch length,
l = half the length of the shaft in inches.

The mass of the shaft section is equal to its weight divided by the constant of gravity, but in this case, as the units used must be the same throughout and as the shaft diameter and shaft length are expressed in inches, we must express the

acceleration of gravity in inches per second per second, instead of feet per second per second. Therefore

$$g = 32.16 \times 12 = 386.$$

From the equation for ω the critical speed of any shaft can be calculated, but this equation is in a rather inconvenient form; it would be much preferable if the critical speed n in revolutions per minute could be calculated directly from the dimensions of the shaft, and this can be done if the equation is suitably transformed.

We have

$$\omega = \frac{2\pi n}{60}, \qquad \omega^2 = \frac{4\pi^2 n^2}{3600},$$

$$I = \frac{\pi d^4}{64}, \qquad E = 30{,}000{,}000, \qquad m = \frac{W}{386},$$

$$W = \frac{\pi d^2}{4} \times 0.28 = 0.07\pi d^2.$$

Hence

$$m = \frac{0.07\pi d^2}{386} = \frac{\pi d^2}{5500},$$

$$l = \frac{L}{2}, \qquad l^4 = \frac{L^4}{16}$$

L being the whole length of the shaft between supports.

Inserting these values in the equation for ω^2 we get

$$\frac{4\pi^2 n^2}{3600} = \frac{\pi^4}{16} \times \frac{d^4}{64} \times 30{,}000{,}000 \times \frac{5500}{\pi d^2} \times \frac{16}{L^4},$$

which when simplified gives

$$n^2 = 2{,}320{,}000{,}000{,}000 \frac{\pi^2 d^2}{L^4}.$$

Therefore

$$n = 1{,}520{,}000 \frac{\pi d}{L^2} = 4{,}800{,}000 \frac{d}{L^2}.$$

This equation applies to solid round shafts. Equations for shafts with other sections can easily be derived by means of the equation for ω^2. It will be seen that this value varies directly as the moment of inertia of the section and inversely as the mass per unit length, all the other factors in the equation being independent of the section. Therefore

$$\omega^2 \sim \frac{I}{m} \quad \text{and} \quad \omega \sim \sqrt{\frac{I}{m}}.$$

But m, the mass per unit length, is directly proportional to the area of the section, which we may denote by A. Consequently,

$$\omega \sim \sqrt{\frac{I}{A}},$$

or as the least radius of gyration of the shaft section. For a solid circle of diameter d

$$\sqrt{\frac{I}{A}} = \sqrt{\frac{\pi d^4}{64} \times \frac{4}{\pi d^2}} = \sqrt{\frac{d^2}{16}} = \frac{d}{4}.$$

For a hollow circle of outside diameter d and inside diameter d_1,

$$\sqrt{\frac{I}{A}} = \sqrt{\frac{\pi(d^4 - {d_1}^4)}{64} \times \frac{4}{\pi(d^2 - {d_1}^2)}} = \sqrt{\frac{d^2 + {d_1}^2}{16}},$$

Hence a tubular shaft of outside diameter d and inside diameter d_1 has a higher critical speed than a solid round shaft of diameter d, the ratio between the two critical speeds being

$$\frac{\sqrt{d^2 + {d_1}^2}}{d},$$

so that the expression for the critical speed of a round tubular shaft becomes

$$n_c = 4{,}800{,}000 \frac{\sqrt{d^2 + {d_1}^2}}{L^2}.$$

Agreement with Practical Observations—It has been found in practice that the actual critical speed is always somewhat lower than the calculated value. For instance, Stodola in "The Steam Turbine" gives several examples of tests for critical speeds of shafts. In five of these tests the critical speeds were found to be 6 per cent, 8 per cent, 9 per cent, 13 per cent and 14 per cent below the calculated values. This discrepancy is undoubtedly due to the fact that the points of support are not rigid. An automobile propeller shaft when running at high speed will whirl in the same way as a heavy rope which is being swung around by two persons. If they cease their whirling effort their hands will nevertheless be carried around in a circle, and so with the propeller shaft supports. The latter consist of the universal joints

which are fitted to shafts overhanging their bearings, and under the influence of the centrifugal force on the propeller shaft these short shafts will bend in the same plane as the propeller shaft, thus virtually increasing the distance between supports. The effect depends, of course, upon the relative stiffness and amount of overhang of the connected shafts, but it has been found that if the calculated critical speed is not approached closer than within 15 per cent a sufficient degree of safety is allowed in ordinary constructions.

The critical speed above discussed is the lowest critical speed. There is an endless number of higher critical speeds, but these are of no interest from a practical standpoint, as the shaft, to be safe, must be made of such dimensions that its lowest critical speed is never attained in practice.

In determining whether there is danger of a design giving trouble from whirling, the maximum possible speed must be carefully determined. This in some cases is much greater than the maximum engine speed. For instance, a motor truck may have a large rear axle reduction and in consequence may have a low maximum speed of perhaps 15 or 20 m.p.h. under power. But this same truck with the clutch out, or the gear in neutral, may be coasted down hill at 40 m.p.h. or more, and the propeller shaft rotate correspondingly rapidly. Tubular propeller shafts are now very generally employed. In the case of very long chassis with unit powerplants where even this expedient is insufficient to prevent whirling, the propeller shaft is made in two parts, connected by a universal joint, the forward part or a short intermediate shaft being supported in an anti-friction bearing mounted on a frame cross member.

Sliding Joint—Owing to the play of the chassis springs, the distance between the shafts connected by the propeller shaft varies continually. If the angularity of the propeller shaft is very slight, this change in length will not be great and certain types of universal joints will compensate for it. However, with the ordinary yoke type of universal joint it is necessary to employ a slip joint. One of the yokes of one of the universal joints is then connected by a sliding-fit spline joint with its shaft, and provisions are made for the lubrication of this slip joint and for excluding dust. If the lubrication of this joint should become defective, the frictional resistance to sliding would be very great, and great end thrust would be created that would have to be taken on bearings at opposite ends of the shaft.

With hollow propeller shafts short solid ends are commonly used, which are united to the tubular sections by weld-

ing. Owing to the fact that commercial seamless tubing cannot be furnished to close limits of concentricity, propeller shaft tubes are now rolled up of sheet steel and welded.

Universal Joints—A universal joint is a joint by means of which it is possible to transmit power or motion between shafts whose axes lie in the same plane and make an angle with each other which may vary while the assembly is in operation. There are some forms of joint which, in addition to permitting the angle between shaft axes to vary in service, permit the axes to be slightly out of line, but these are not generally required for automotive transmissions. Where the change speed gear is mounted separately on the chassis frame it is customary to have a short intermediate shaft between it and the engine, which shaft is fitted with a universal joint at each end. This is necessary because of the lack of rigidity

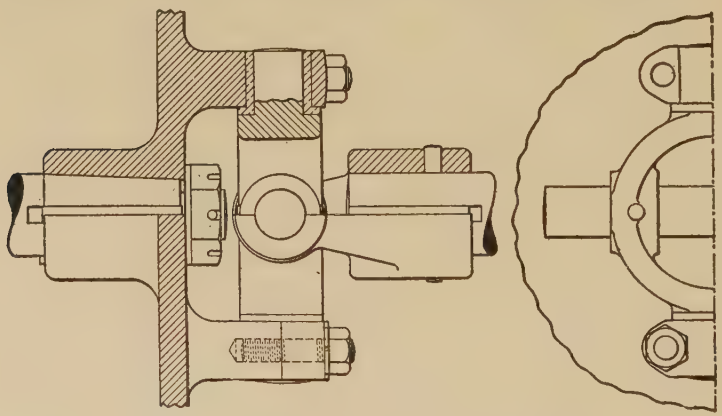


FIG. 3.—INTERNAL-RING TYPE OF JOINT

of the frame, and the consequent impossibility of keeping the transmission drive shaft in alignment with the engine crankshaft. The most important application of universal joints in automotive vehicles, however, is in the final drive from the transmission to the rear axle. Every shaft-driven car has at least one universal joint in the propeller shaft, and a great many have two.

The ordinary metal type of universal joint comprises two yokes and a member connecting these yokes which is in the form of either a cross or a ring. In the original cross-type of universal joint two pins passed through the cross or central block, with their axes a small distance apart so as to avoid interference, these pins having bearings in eyes on the yokes.

It is desirable, however, to have the axes of the two pins intersect, and a slightly modified design which embodies this feature is illustrated in Fig. 3. This particular joint is designed for use directly behind the gearcase. One of the yokes is integral with the brake drum center or flange, and the two yokes are connected by a ring which has two trunnions formed integral with it, while a pin extends through the ring and through eyes on the yoke of the driven shaft.

In the type of joint shown in Fig. 4 the two yokes are formed with trunnions on their arms and these trunnions are supported in bearings formed between the halves of an external ring, which latter is held together by eight bolts.

Periodical Speed Fluctuations—A feature of the universal joints described above is that they do not transmit motion uniformly when the shafts are at an angle with each other;

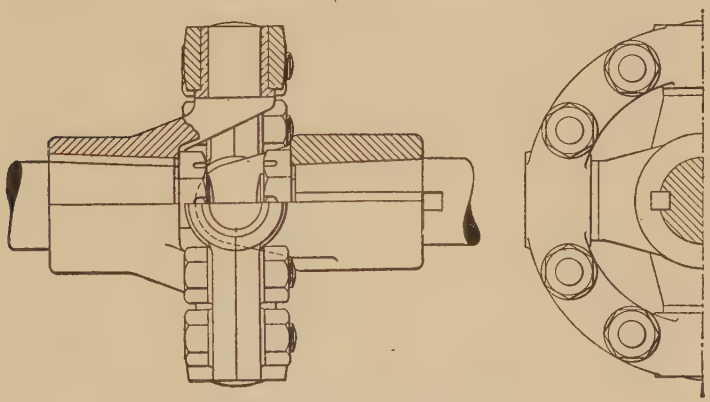


FIG. 4.—SPLIT-RING-TYPE OF UNIVERSAL JOINT.

that is to say, if the driving shaft turns at uniform speed, the speed of the driven shaft will vary periodically, being soon less and soon greater than the speed of the driving shaft. The common feature of all of these joints is that they have two rocking axes at right angles to each other.

To gain an idea of the magnitude of the variation in angular velocity, we will assume a universal joint connecting two shafts in a vertical plane, the driving shaft being placed horizontally, as in Fig. 5. The axes of the two pins intersect each other, their ends being designated by AA and BB, respectively. When the joint is in motion the line AA describes a circle in a vertical plane, and the line BB a circle in a plane making with the vertical an angle ϕ, equal to the angle

between the two shafts. These two circles are great circles of the same sphere, the common diameter being a line through the point C perpendicular to the paper. Points A and B always remain at the same distance from each other, viz., one quadrant of a great circle. The deviation in the direction of travel is the greatest when either point A or point B coincides with the points of intersection of the great circles. When the points A coincide with these points of intersection, the angular speed of the driven shaft is smaller than the angular speed of the driving shaft, and when points B coincide with these points of intersection the angular speed of the driven shaft is greater than the angular speed of the driving shaft. There are four points in each revolution in which driving and driven shafts rotate at equal angular speeds, these being

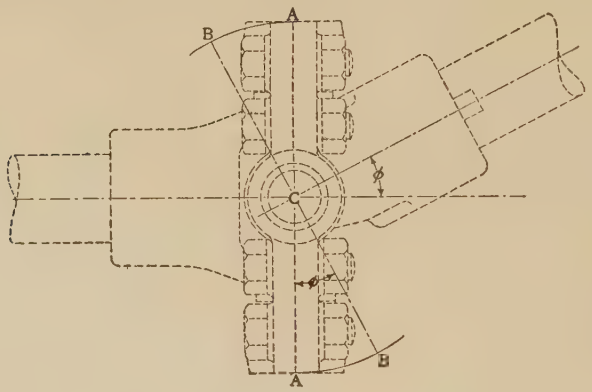


Fig. 5.

located substantially midway between the points of maximum and minimum speeds of the driven shaft.

Let the two large arcs in Fig. 6 represent the great circles in which the points A and B travel. Let point A travel from the point of intersection to point A' and point B travel at the same time to B', which is determined by the fact that A' B' must be a quadrant. Now, lay off from the point B' on the line of travel of point B a quadrant, or 90°, which will give point C. Through A' and C draw an arc of a great circle. Angles B' A' C and B' C A' are both right angles (because their opposite sides are quadrants), hence angle A C A' is a right angle. We, therefore, have a right-angled spherical triangle A A' C, the angle A' A C of which is equal to the angle between the two connected shafts, the side AA'

of which represents the angular motion of the driving shaft, and the side $A\ C$ the angular motion of the driven shaft during a short period after the point A has passed through the point of intersection; in other words, when the pin of the driving shaft is at right angles to the plane through the two connected shafts.

According to a theorem of spherical trigonometry,

$$\cos A'AC = \tan AC \cot AA'$$

Since the tangent is the reciprocal of the cotangent we may write this

$$\frac{\tan AC}{\tan AA'} = \cos A'AC.$$

and since for very small angles the tangents are proportional to their angles, we have

$$\frac{AC}{AA'} = \cos A'AC,$$

Therefore, when the pin of the driving shaft is perpendicular to the plane through the connected shafts, the angular velocity of the driven shaft is smaller than the angular velocity of the driving shaft in the proportion of the cosine of the angle between the two shafts to unity. It may be shown in a similar way that when the pin on the driving shaft is in the plane of the two connected shafts the driven shaft runs faster than the driving shaft in the inverse proportion.

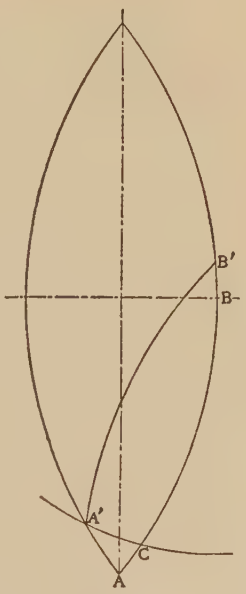


FIG. 6.

It is also of interest to find an expression for the momentary ratio of angular velocities at any point in the revolution of the driving shaft. To simplify the expressions, we will denote the angle $A'\ A\ C$ by ϕ, the side $A\ A'$ by a and the side $A\ C$ by b. Then we have as before

$$\tan b = \cos \phi \tan a.$$

Differentiating, we have

$$\sec^2 b\ db = \cos \phi \sec^2 a\ da.$$

and

$$\frac{db}{da} = \cos\phi \frac{\sec^2 a}{\sec^2 b},$$

which gives the ratio of angular velocities at any moment in terms of ϕ, a and b. It is preferable, however, to express the value in terms of ϕ and a only, as b is not directly known, and the latter can be easily eliminated. Squaring the equation for tan b we have

$$\tan^2 b = \cos^2 \phi \tan^2 a.$$

Adding 1 to each side of the equation—

$$1 + \tan^2 b = 1 + \cos^2 \phi \tan^2 a.$$

But since

$$1 + \tan^2 b = \sec^2 b,$$

we may substitute the right hand term of the last equation for $\sec^2 b$ in the equation for db/da,

$$\frac{db}{da} = \cos\phi \frac{\sec^2 a}{1 + \cos^2 \phi \tan^2 a},$$

which gives the ratio of angular velocities after any angular motion a of the driving shaft from the zero position, in which latter the pin of the driving fork is perpendicular to the plane through the two shafts. When $a = 0$ the last equation reduces to

$$\frac{db}{da} = \cos\phi.$$

which is the same as already found for the position of minimum speed of the driven shaft.

The curve, Fig. 7, shows the variation in speed of the driven shaft during a motion of one-half a revolution or 180°, the driving shaft making 1000 r.p.m. and the angle between the shafts (ϕ) being 30°. We start with the position where the pin of the driving yoke is perpendicular to the plane of the two shafts. In this position the driven shaft rotates at the rate of 866 r.p.m., its lowest speed. The speed of the driven shaft increases until after a little more than 45° motion of the driving shaft it equals the speed of the latter. It keeps on increasing, and after 90° motion, when the pin of the driving yoke is in the plane of the connected shafts, it attains its maximum speed of 1155 r.p.m. Then it decreases again, according to the same curve, until after 180°, or one-half revolution, it again attains its minimum speed of 866 r.p.m. During one revolution the speed of the driven

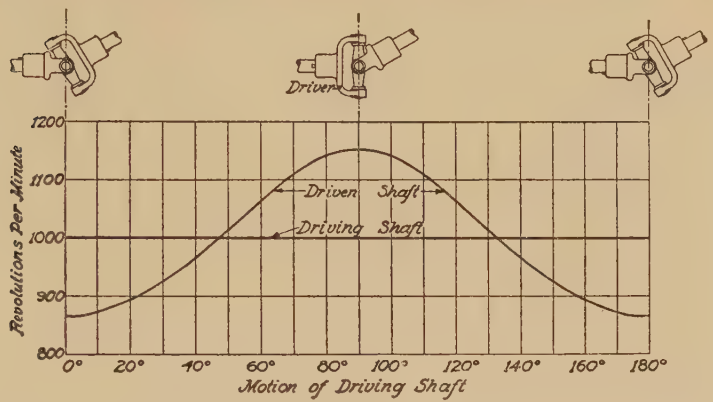


FIG. 7.—VARIATION OF DRIVEN-SHAFT SPEED (ANGLE BETWEEN SHAFTS, 30°).

shaft, therefore, passes through two maxima and two minima. Its average speed, of course, is the same as that of the driving shaft, and the speed fluctuation amounts to

$$\frac{(1155 - 866) \times 100}{1000} = 28.9 \text{ per cent.}$$

The following table gives the speed fluctuations in the driven shaft corresponding to different angles between shafts, the speed of the driving shaft being assumed to be constant:

Angle ϕ Degrees	Fluctuation (Per Cent)	Angle ϕ (Degrees)	Fluctuation (Per Cent)
2	0.15	16	7.9
4	0.5	18	10
6	1.1	20	12.4
8	2	22	15
10	3	24	18
12	4.4	26	21.3
14	6.	28	25

This fluctuation in the speed of transmission is a matter of great moment. In a gasoline automobile we have at one end of the transmission line the engine, whose speed is maintained substantially constant by a heavy flywheel, and at the other end the car, which, when running at high speed, also has its speed maintained by inertia. But if the transmission is effected through a single universal joint working at an

appreciable angle, the speed of either the car or the engine, or of both, must of necessity change greatly in a quarter revolution of the driving shaft. The flywheel inertia strongly resists such a change in the speed of the engine, and the car inertia a change in the speed of the car, and the result is that every part of the transmission line is subjected to enormous stresses. Not the least to suffer under these stresses are the tires, which tend to slip on the ground as the car tends to suddenly accelerate. To minimize these stresses the drive must be so arranged that the two shafts are always nearly in line with each other. They can be entirely eliminated by using two universal joints in series. We found that the speed of transmission is reduced in a certain ratio when the pin of the driving fork is perpendicular to the plane through the shafts, and increased in the inverse proportion when the pin of the driving shaft is in the plane of the shafts. These two positions are 90° apart. Hence, by arranging two universal joints in series in such relation that the driving yoke or corresponding member of the second is set at an angle of 90° with respect to the driving yoke of the first, and so that the driving and driven shafts make equal angles with the intermediate shaft, motion will be uniformly transmitted from the driving to the driven shaft. There are two arrangements that meet these conditions, and these are illustrated in Fig. 8. In the first the driving and driven shafts are maintained parallel with each other. This is readily accomplished in a car provided with the so-called Hotchkiss drive in which the only connection between the driving axle and the frame is through half-elliptic springs. If the springs are symmetrical, the axle moves in a plane perpendicular to that of the frame, and the rear axle pinion shaft can be maintained parallel to the frame and therefore to the transmission tailshaft. With the other arrangement the driving and driven shafts both deviate toward the same side from the propeller shaft and must make equal angles with it. This can be approximated by using a torque arm on the rear axle which is of about half the length of the propeller shaft. With both arrangements the yokes at opposite ends of the propeller shaft must lie in the same plane.

Details of Design—The major parts of metal universal joints are drop-forged of medium carbon steel and heat treated, while the bearing bushings are made of hardened steel. Using hardened steel rather than bronze bushings permits of using higher unit bearing pressures and of reducing the dimensions of the joint for a given capacity.

The capacity of a joint is a rather uncertain factor. It

evidently depends to some extent upon the average power factor of the engine in the service for which it is intended, and to an even greater degree upon the angle at which the joint must work. If the shafts connected are very nearly in line there is very little motion at the bearings, and heat generation and power losses are small, whereas the reverse is the case when the joint works at large angles. Finally, the low gear reduction may have some influence, as the joint must be capable of transmitting the full engine power when operating in low gear.

For joints of similar design the torque capacity evidently varies as the cube of the linear dimensions, because the bearing area varies as the square of the linear dimensions and the permissible bearing load the same. The force represented by this load acts through an arm whose length is proportional to the linear dimensions. For use on passenger cars, and where the angularity of the joint is moderate, the torque

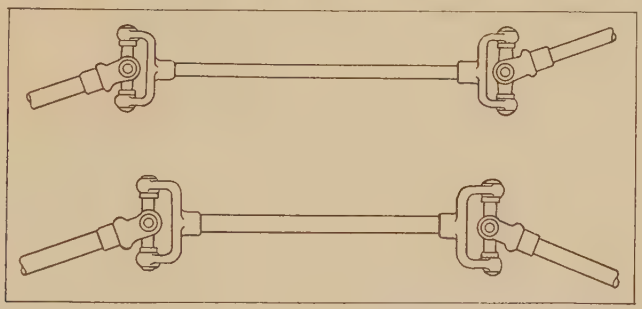


FIG. 8.—TWO ARRANGEMENTS OF UNIVERSAL JOINTS ENSURING UNIFORM TRANSMISSION OF MOTION.

capacity of a conventional metal universal joint is given approximately by the equation

$$T = 30\, r^3 \text{ lb.-ft.},$$

where r is the mean radius of the trunnion bearing surface (relative to the axis of the joint) in inches. The unit pressure on the projected bearing surface for maximum torque on direct drive is made from 1000 to 1200 lbs. per sq. in., the average value being 1100 lbs. If the trunnions have a length equal to their diameter the trunnion diameter figures out to an average of 0.574 r.

Companion Flange—With certain types of universal joint in which the trunnions are integral with a cross, block or ring, assembly and disassembly would be very difficult if the

yokes were secured directly to the transmission tailshaft and the rear axle drive pinion shaft. This difficulty is overcome by making use of a companion flange, which is keyed or splined to the shaft, one of the yokes of the joint being made with a similar flange and the two flanges bolted together (see Fig. 10). When the bolts are removed the universal joint can be moved sidewise and slipped off the propeller shaft.

Pot-Type of Joint—In Europe wide use has been made of what is known as the pot-type of universal joint, illustrated in Fig. 9. It comprises a bell-shaped steel forging with thick walls, that is secured to one of the shafts to be connected and has two diametrically opposite longitudinal slots milled in its wall. The other shaft is formed with a ball-shaped end that is cross-drilled for a pin which extends into the slots in the shell. Hardened steel trunnion blocks are interposed between the pin and the surfaces of the slot, to distribute the bearing pressure. This type of joint can be readily en-

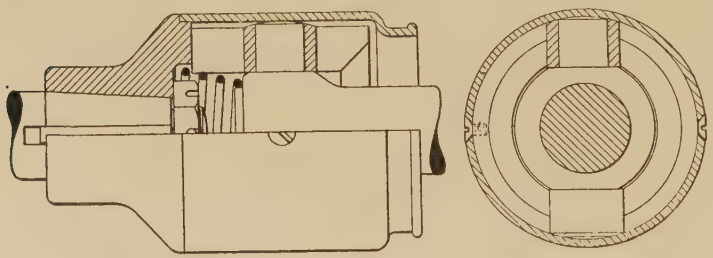


FIG. 9.—POT-TYPE OF UNIVERSAL JOINT.

closed for grease lubrication, by means of a leather boot extending between a groove at the open end of the shell and a section of the shaft outside the shell, and it has the further advantage that it serves also as a slip joint. But it is slightly less efficient than the type in which there is rocking motion around two pairs of trunnions at right angles to each other.

Comparison of Pot- and Cross-Types of Joint—According to the modern theory of lubrication, the friction of well-lubricated bearings is proportional to the product of area of oil shear and velocity of shearing motion, and the frictional loss is proportional to the friction and the distance through which it must be overcome. The velocity of shearing motion, of course, is directly proportional to the distance of the motion at the sliding surfaces during a cycle or half a cycle.

We found above that the trunnion diameter d is equal to 0.574 r, where r is the mean radius of the trunnion block

bearing surface, and it is therefore equal to 0.287 D, where D is the mean diameter of the circles described by points in the trunnion block bearing surfaces. Assuming the trunnions to have a length equal to the diameter d, and the trunnion blocks to have a length of face 1.25 d and a width d, the ratio of shearing areas will be

$$\frac{\text{trunnion}}{\text{trunnion block}} = \frac{3.14\,d^2}{2 \times 1.25\,d^2} = 1.256.$$

The ratio of motions will be

$$\frac{\text{trunnion}}{\text{trunnion block}} = 0.287.$$

Hence the ratio of frictional forces will be

$$\frac{\text{trunnion}}{\text{trunnion block}} = 1.256 \times 0.287 = 0.36$$

and the ratio of frictional losses,

$$\frac{\text{trunnion}}{\text{trunnion block}} = 0.36 \times 0.287 = 0.103.$$

Since one joint has two pairs of trunnions and the other one pair of trunnions and one pair of blocks, the frictional losses of the two joints should bear to each other the proportion

$$\frac{\text{cross type}}{\text{pot type}} = (0.103 + 0.103)/(1 + 0.103) = 0.187.$$

In other words, for the same angularity and the same speed of revolution the losses in the cross type are only a little more than one-fifth as great as in the pot type (under the conditions assumed).

Lubrication—Effective lubrication of universal joints operating at high speeds is a difficult problem, as the centrifugal force has a tendency to cause the lubricant to flow away from the bearing surfaces. Most of the inventive effort expended on universal joints since the beginning of the automobile era has had for its object improvement of the lubrication of the device. Fig. 10 shows the Spicer joint, which is grease-lubricated. A pressed steel inner housing comprising a spherical zone concentric with the joint is bolted to the flange yoke, while an outer pressed steel housing, also in the form of a spherical zone, has a hub by which it is clamped to the hub of the plain yoke. Near the outer edge of this latter housing there is a groove which takes a felt packing

that bears against the outer surface of the inner housing, thus ensuring a grease-tight joint in spite of the relative movement of the two housings. The inner housing is provided with a plug-closed opening through which grease can be injected by means of a gun. A detail of the Spicer joint worth mentioning is that the eyes of the yokes are open and the bushings are held against centrifugal force by spring steel rings forced into grooves provided for the purpose.

Oil-Lubricated Joints—Fig. 11 shows the Blood Brothers universal joint, which is oil-lubricated. It comprises a central block through which pass two pins at right angles to each other, one of these being of plain cylindrical form and the other shouldered. The ends of these two pins form trunnions having their bearings in the yokes. Bearing bushings

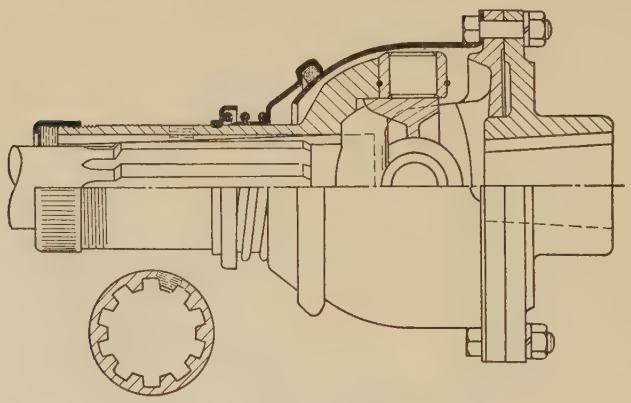


FIG. 10.—GREASE-LUBRICATED UNIVERSAL JOINT (Spicer).

with knurled outer surfaces are pressed into the eyes of the yokes and are secured against endwise motion by peening over the metal of the yokes. To the outer, projecting ends of the bushings are secured caps. Both cross pins are drilled through their center, and the trunnion portions are counterdrilled to get sufficient oil space. A cork gasket is provided at the inner end of the bushing, and a felt gasket under the flange of the cap, to ensure oil-tightness.

Another design of oil-lubricated universal joint, shown in Fig. 12, is made by the Mechanics Machine Company, Rockford, Illinois. It comprises an annular ring or housing made of pressed steel, in halves, with ground joint surfaces and held together by four studs with nuts, these studs being en-

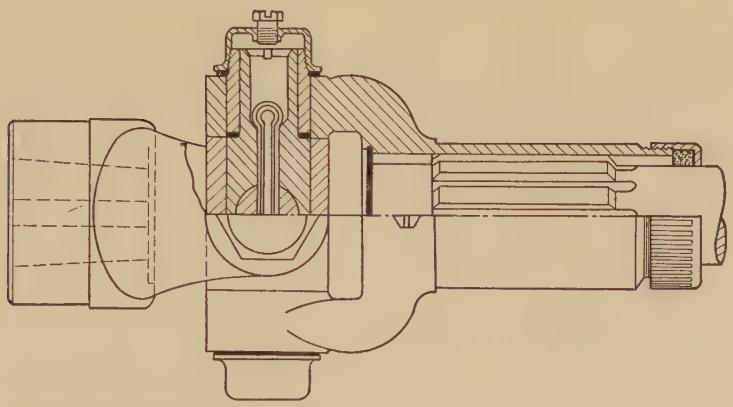


FIG. 11.—CROSS-TYPE OF UNIVERSAL JOINT (Blood Brothers)

larged at their center and forming bearings for the trunnions of the yokes. Felt washers are provided where the trunnions enter the housing, and the nuts on the clamping studs are locked by stampings which pass over two adjacent studs and have their ends turned over. The housing is filled with oil through a plug hole.

Fig. 13 shows still another oil-lubricated joint, the Cleveland, which comprises a central cup-shaped member with four radial trunnions, and two two-armed spiders, into which are fastened bearings for the trunnions. These bearings are closed at their outer ends. The cup is closed by a sheet-metal cover to form an oil reservoir, and the trunnions as well as

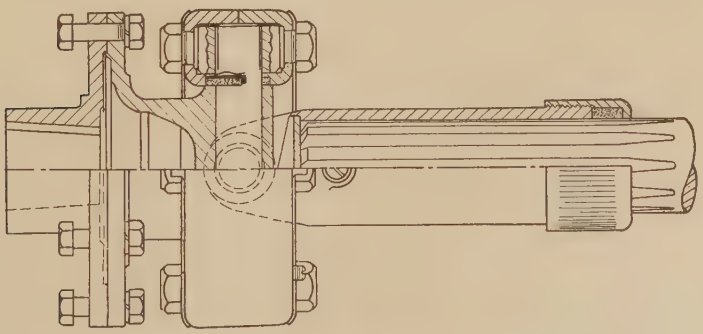


FIG. 12.—OIL-LUBRICATED JOINT (Mechanics Machine Co.)

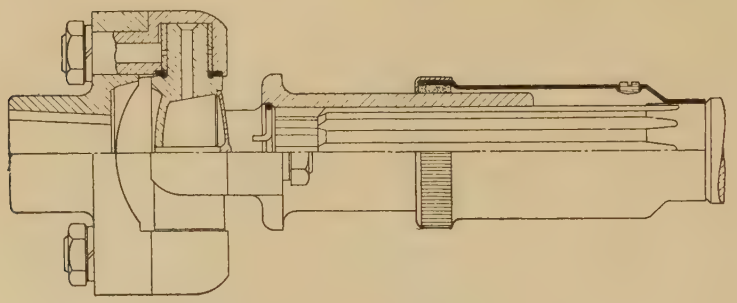


FIG. 13.—CLEVELAND UNIVERSAL JOINT.

the studs which support the bearings, are drilled out radially to form oil pockets and passages.

Weiss Constant Velocity Joint—Carl W. Weiss of New York has invented a universal joint (Fig. 14) which gives a constant velocity ratio between the driving and driven shafts. It consists of two spiders with two, three or four arms each, each arm of one of the spiders being located between two adjacent arms of the other, and separated therefrom by steel balls located in grooves in the sides of the arms. The elements of the surfaces of these grooves are straight lines, and the grooves can be cut by milling. However, the two grooves holding one ball can never become parallel, and the balls therefore cannot become dislocated. When the connected shafts are in line the balls lie in a plane perpendicular to the common shaft axis. If one of the shafts is then swung around to make an angle with the other, the plane of the balls turns through half that angle. That the transmission of motion is uniform can be readily demonstrated with a model by fitting each of the connected shafts with a long

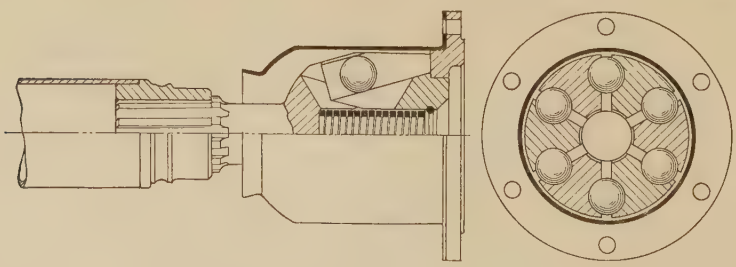


FIG. 14.—CONSTANT-VELOCITY UNIVERSAL JOINT (Weiss).

pointer which moves over a dial. When the shafts are then turned to make an angle with each other, and rotated, both pointers will always indicate the same angular motion. These universal joints, which are particularly adapted for use in connection with front-wheel drives and where only a single joint is used in a rear-wheel drive, can be made to operate at shaft angles up to 35°. If a greater angularity is required, as in front drive buses, two joints are connected together,

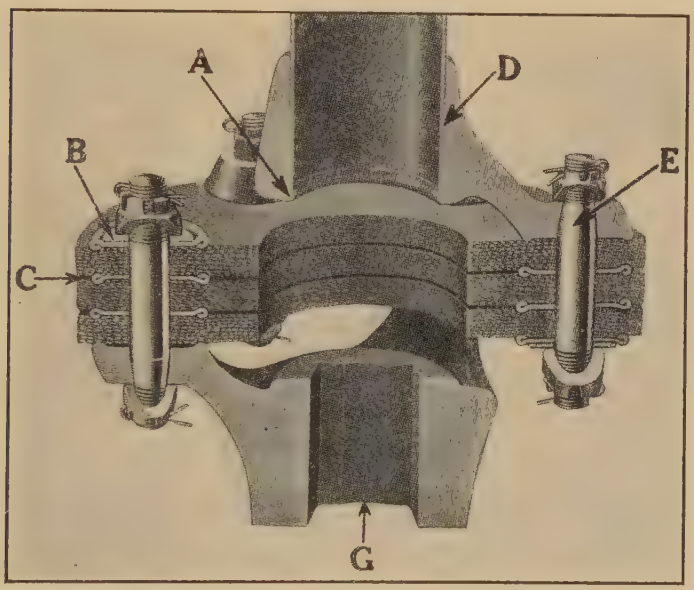


FIG. 15.—SPICER-HARDY FABRIC-DISC TYPE OF UNIVERSAL JOINT.

A, ar-weld of universal-join spider to tubular shaft; B, C, steel washers distributing pressure over surface of discs; D, indicating close fit of spider on tubular axle; E, stud with taper fit in spider arm; G, drop-forged spider.

and motion can then be transmitted uniformly at any angle up to 50°.

Disc-Type Universal Joints—Considerable trouble was experienced with the early metal universal joints owing to the lack of efficient means of lubrication, and flexible steel disc and leather disc joints were proposed and used to a certain

extent, these requiring no lubrication. A considerable advance along this line was achieved in 1911 when Edward J. Hardy in England introduced a special rubberized fabric disc in which the various layers of fabric were so piled up that there was uniform angular spacing between the strands of cotton of successive layers, thus ensuring uniform torsional strength of the disc in all directions. A fabric-type universal joint consists essentially of two spiders, usually three-armed; a number of fabric discs or rings between them, and bolts, nuts and washers by which the fabric rings are secured to the spiders. The form of the washers is important, as upon it depends the distribution of stress in the fabric rings. Fig. 15 shows the Spicer fabric universal joint.

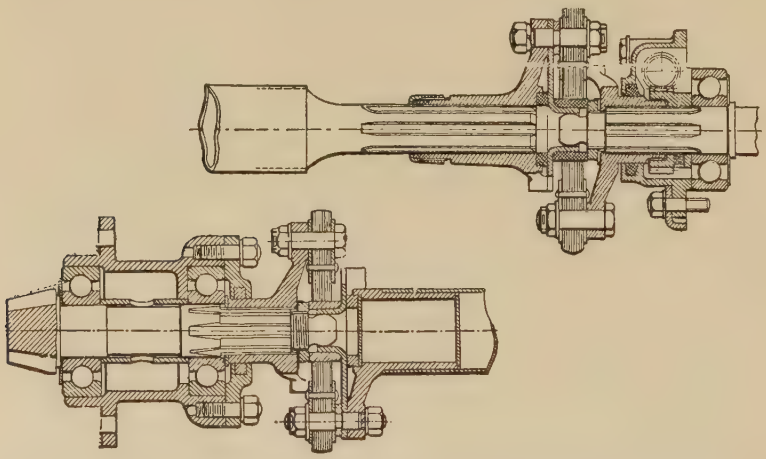


Fig. 16.—Fabric-type Universal Joint with Metal Center-guide.

As compared with metal universal joints the fabric type has the advantages that it remains silent under all conditions and that it requires no lubrication. This type has its limitations, however, and at the present time it is not much used in final drives in passenger cars in the United States, while about 1920 it held an important place in this field. Some trouble was experienced from excessive end thrust, which injured the discs. The chief difficulty, however, seems to have arisen from the inability of the fabric discs to hold the shafts permanently concentric. When a shaft rotates around another than its geometrical axis the tendency toward

whirling is naturally greatly increased, and with increase in engine speeds to above 2500 r.p.m. this became a serious matter. In England fabric disc joints are sometimes provided with a metal support at the center, as shown in Fig. 16, to prevent this whirling. Fabric joints are suitable only for drives in which the angle between connected shafts can be kept small, the limit of angularity being generally placed at 7°. Fig. 16 is reproduced from *Engineering* of London.

Capacity of Disc-Type Joints—The Hardy disc, made of long-strand Sea Island cotton and a good grade of friction rubber compound, is said to have a tensile strength of 3400

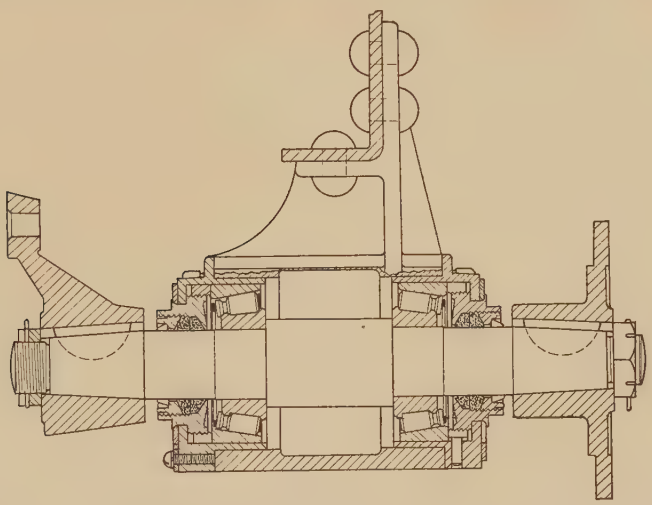


Fig. 17.—Short Intermediate Shaft Supported From Frame Cross-member on Two Roller Bearings.

lbs. per sq. in. In selecting the proper size of joint for any given propeller shaft installation it is usual to figure with a stress in the disc of 500 lbs. per sq. in. under full engine torque through the low gear, figuring with the full section of the disc without allowance for bolt holes. Where there is likely to be appreciable angularity, three-armed spiders are generally employed, while for installations between the clutch and transmission four-armed spiders may be used and will permit of cutting down the required diameter slightly.

Intermediate Bearings for Propeller Shafts—It has been pointed out already that where the distance from the rear of

the gearcase to the rear axle drive pinion is very large, the propeller shaft is made in halves which are joined by a universal joint. The rear end of the forward half is then supported in a bearing mounted in a frame bracket and dimensions for the shaft end and bearing mounting have been standardized by the S. A. E. (see Appendix). A rather different arrangement is used on Diamond T trucks. On these, as shown in Fig. 17, a short shaft mounted in two Timken roller bearings in a frame bracket, is connected by universal joints to both the forward and rear propeller shafts.

CHAPTER VI

The Differential Gear

In its broadest sense the term differential gear describes a device which permits of torque from a single driving shaft being transmitted to two driven shafts for both directions of rotation, while leaving the two driven shafts independent of each other with respect to speed. It is very desirable to apply driving torque to at least two road wheels, in order that sufficient traction may be available under all conditions; and in describing curves it is necessary that one wheel can rotate faster than the other, as otherwise one tire would have to slip over the ground in order to make up for the difference in the lengths of the inner and outer wheel tracks. Moreover, if there were no such differential action and the two wheels were rigidly secured to a common driving axle or shaft, the vehicle would not respond readily to the steering gear, and there would be a strong tendency toward front (steering) wheel skid.

There are two common types of differential gear, the bevel type and the spur type, of which the former is in by far the most extensive use. These conventional differential gears, also called equalizing gears, impress equal torques upon both driven shafts under all conditions. There is another class of differential which, when the vehicle is describing a curve, applies driving torque only to the inner wheel, and when one wheel looses traction, continues to apply torque to the other, in which respect it differs from the conventional type. This special non-slip differential will be referred to at some length in the latter part of this chapter.

A bevel-type differential gear (Figs. 1 and 2) consists of two bevel gears called side gears, arranged coaxially and facing each other, and a varying number of bevel pinions between them and meshing with both. Generally either two or four pinions are used, which are placed at equal angular distances. The pinions are capable of rotating on radial pins which are clamped at their outer ends between the two halves of a

housing. This housing is provided with hubs carried in ball or roller bearings in the rear axle or jackshaft housing and with a flange to which the driven bevel gear, worm gear, etc., can be secured.

Action of the Differential—Power is thus applied to the housing of the differential. The housing transmits it to the bevel pinions, the latter to the bevel gears, and these to the rear axle shafts or jackshafts. Under any given conditions of operation, a certain torque is impressed upon the differential housing. This torque is divided equally between the two or four bevel pinions. Each bevel pinion constitutes a balance lever between the two bevel gears and evenly divides its torque between them. Thus the total torque impressed

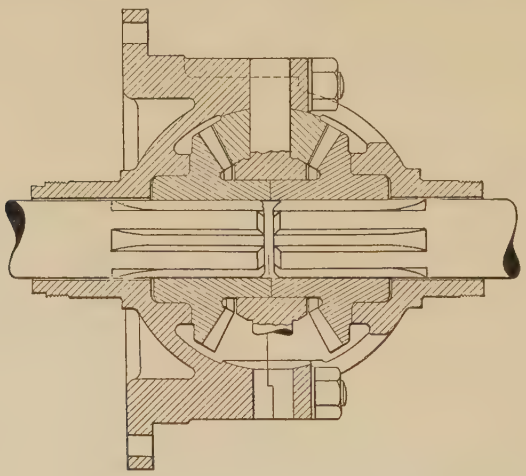


FIG. 1.—LONGITUDINAL SECTION THROUGH BEVEL-TYPE DIFFERENTIAL GEAR.

upon the differential housing is at all times equally divided between the two side gears.

The relative motion of the two side gears depends upon the position of the steering gear and upon the traction conditions. Suppose first that both driving wheels run on dry road surface so there is plenty of road adherence. Then the rate of revolution of each wheel and that of the corresponding side gear of the differential will depend upon the length of the path followed by that wheel. If the steering wheels are in the straight-ahead position and both driving wheels have exactly the same diameter, then both will rotate at the same

speed, as will the differential side gears. On the other hand, if the steering wheels are deflected from the straight-ahead position, the vehicle is constrained to travel in a curve, and the wheels on the outside of the curve will be forced to turn faster than those on the inside. Under these conditions the pinions of the differential will turn on their pins, allowing one side gear to run faster than the other. The speed of the housing of the differential is always equal to the algebraic mean of the speeds of the two side gears.

In case one of the wheels stands on slippery ground and has insufficient road adherence, it will slip. The differential gear under these conditions also divides the propelling effort equally between the two driving wheels, and the wheel which stands on dry surface can exert no more propelling effort

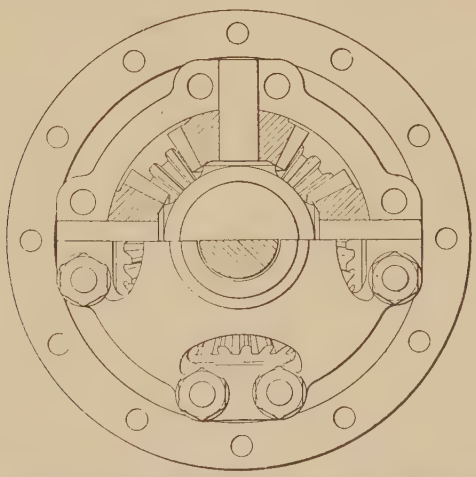


FIG. 2.—HALF-SECTIONED VIEW OF BEVEL-TYPE DIFFERENTIAL GEAR.

than the one on slippery surface. The car will thus be stalled, and the wheel on slippery ground will be spun around at twice the rate at which it would otherwise turn with the engine running at the same speed, whereas the other wheel will remain stationary. This quality may be regarded as a defect of the differential gear, especially in the case of very heavy vehicles, and such vehicles sometimes are provided with a differential lock, consisting of some means for connecting the two side gears together so that they must rotate in unison.

The gears and pinions of differentials are generally made of a case-hardening alloy steel (No. 2320 S. A. E.), and then

combine great surface hardness with very fair tensile properties of the core. The two weakest elements in a bevel differential are generally the pin surfaces on which the pinions bear and the teeth of the pinions. Calculation of these elements should be based upon the maximum rear axle torque with the engine driving through the low gear of the transmission. Let the maximum rear axle torque (maximum engine torque $\times$ transmission low-gear ratio $\times$ rear axle reduction ratio) be designated by T; let the mean distance of the pin bearing surfaces from the axis of the differential be r, and let there be n pinions of a diameter d and length of pinion bearing l. Then the load on each pinion bearing is

$$P = \frac{T \times 12}{r \times n},$$

and the unit load on the pin bearings,

$$p = \frac{T \times 12}{rndl}.$$

The value of p in automotive differentials varies between 4000 and 5000 lbs. per sq. in., 4500 lbs. being a good average. The higher value mentioned should be used only if the reduction ratio of the lowest gear in the transmission is unusually large and is intended only for use in emergencies, while the lower value can be used in high-class constructions where it is desired to minimize wear and depreciation.

Strength of Bevel Gears—Bevel gears can be calculated for strength by an adaptation of the Lewis formula taking account of the variation of the circular pitch and of the pitch radius along the face.

The bevel gear tooth decreases in pitch uniformly from the outer end, where the pitch has the nominal value, to the apex of the gear cone. Let the face width be equal to $1 - \alpha$ per cent of the cone distance, which distance we may designate by L. Now let us take any small section dx of the tooth at a distance x from the apex (Fig. 3). The tangential pressure which this section will support we know to be proportional to the circular pitch and to the width of face dx. But the circular pitch at this part of the face width, if p is the nominal circular pitch, is

$$p_x = \frac{x}{L} p.$$

Also, a tangential force F_x on the pitch circle at a distance x

from the apex is equivalent to a tangential force on the maximum pitch circle

$$dF = F_x \frac{x}{L}.$$

Hence we may write for the tangential force which the section dx of the tooth will support (using the Lewis formula)

$$dF_x = S \times \frac{x}{L} p \times y \times dx = \frac{Spy}{L} x\,dx,$$

and this is equivalent to a tangential force on the pitch circle of the large end of

$$dF = \frac{Spy}{L} x^2 dx \times \frac{x}{L} = \frac{Spy}{L^2} x^2 dx.$$

If we integrate this expression between the limits $x = L$ and $x = a\,L$ we get

$$F = \int_{aL}^{L} \frac{Spy}{L^2} x^2 dx = \frac{Spy}{L^2}\left(\frac{L^3}{3} - \frac{a^3 L^3}{3}\right) = \frac{SLpy}{3}(1 - a^3).$$

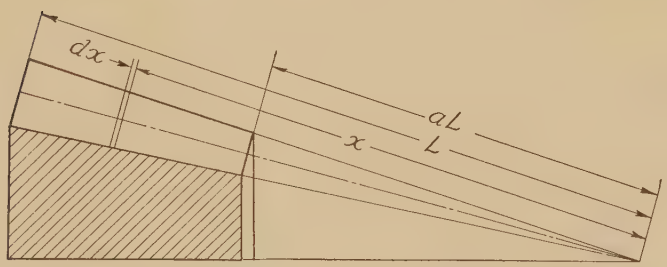


FIG. 3.—BEVEL-GEAR TOOTH-STRENGTH DIAGRAM.

in which F is the tangential force on the pitch circle at the large end; S, the permissible stress in pounds per square inch; p, the circular pitch at the large end; L, the cone distance; y, the Lewis constant for the particular number of teeth, and a, the proportion of the pitch line length from the inner end of the pinion to the apex, to the cone distance.

The Lewis formula in general errs on the safe side, in that it assumes that all of the tooth pressure comes on a single tooth and at the very top thereof, whereas in practice, with accurately spaced gears, the tooth load is divided between two teeth part of the time, and when it is taken by a

single tooth the contact is near the pitch line, which corresponds to a greatly reduced bending stress. Slight errors in spacing and contour cannot be entirely avoided, but up to a certain extent these are compensated for by the elasticity of the gear material.

Owing to this characteristic of the Lewis formula, the unit stress which can be used in it when it is applied to the elements of differential gears is very high. Another circumstance also favoring the use of high unit stresses in the calculations is that most of the time the gears and pinions of the differential are under static loads only and when they do move, their speeds are low, hence the load increment due to speed is not an important factor.

The tensile stress in the teeth of the differential pinion, calculated by means of the modified Lewis formula as given above, should be between 80,000 and 120,000 lbs. per sq. in., 100,000 being a good average figure. What was said in regard to the use of the upper and lower values of the unit pressure on the pins applies also in this case. For a first approximation in determining the required pitch diameter of the side gear for a four-pinion differential use can be made of the equation

$$D = 0.275 \sqrt[3]{T},$$

and for a two-pinion differential

$$D = 0.300 \sqrt[3]{T},$$

where D is in inches and T is the rear axle maximum torque in lb.-ft.

Proportions of Gears—Since the strength of gears increases with the circular pitch, and smooth tooth action is not so essential, on account of the low speed or lack of speed of these gears, a coarse-pitch tooth is generally used. In passenger car work the pinions are made with from 10 to 12 teeth and the gears with from 15 to 22. There is no theoretical reason why larger numbers of teeth should be used for the differentials of heavier commercial vehicles with greater rear axle torque loads, but there is a certain convenience in using the same pitch for various sizes, and actually the increased capacity of the larger differentials is obtained partly by increasing the number of teeth and partly by increasing the circular pitch. Thus the differential of a certain 5-ton truck has 15 and 30 teeth and that of a moderate-sized, high-speed bus, 12 and 26 teeth.

Center of Load Distribution—Bevel gears produce both radial loads and thrust loads on the bearings supporting them. Before these bearing loads can be calculated, it is necessary to determine the center of load distribution along the face of the gear. We will assume that the tooth load is distributed along the tooth in proportion to the strengh of the tooth section, which is an ideal condition. In practice, of course, much depends upon the accuracy of the gears and upon their mounting and adjustment. Let n be the ratio of the tooth face to the cone distance and m, the proportion of the cone distance represented by the distance from the apex to the center of the tooth load. From the well-known formula for the strength of gears we know that the strength of a tooth section is directly proportional to the circular pitch at that section, and the pitch, of course, decreases uniformly from the large end of the tooth to the apex, where it is zero. The strength of the section of a bevel gear is also proportional to the width of that section. In Fig. 4 the vertical lines *ab*,

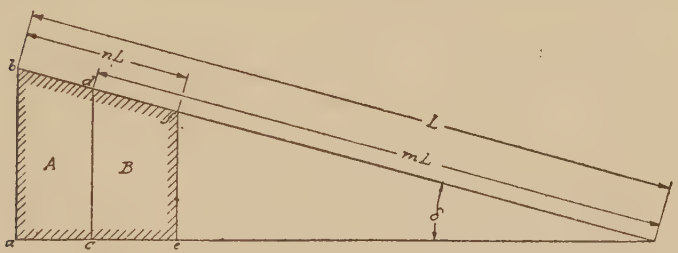


FIG. 4.—CENTER-OF-LOAD-DISTRIBUTION DIAGRAM.

cd and *ef* represent the circular pitch at the respective points and the line *cd* is supposed to divide the entire gear into two parts of equal strength. As the strength of a gear is proportional to the product of its circular pitch into its width of face, the two areas A and B should be equal. If this is the case, then

$$\left[\frac{L \sin \delta + L(1-n) \sin \delta}{2}\right] n L \cos \delta = \left[\frac{L \sin \delta + mL \sin \delta}{2}\right] (L - mL) \cos \delta \times 2$$

$$\frac{L + L(1-n)}{2} nL = (L + mL)(L - nL)$$

$$\frac{2L - nL}{2} nL = L^2 - m^2L^2$$

$$nL^2 - \frac{n^2L^2}{2} = L^2 - m^2L^2$$

$$n - \frac{n^2}{2} = 1 - m^2$$

$$m = \sqrt{1 + \frac{n^2}{2} - n}.$$

For convenience in making calculations, the values of m for a number of values of n within the usual range are tabulated below.

n.....	0.24	0.26	0.28	0.30	0.32	0.34	0.36
m.....	0.888	0.879	0.871	0.863	0.855	0.847	0.840

Bevel Gear Bearing Loads—We are now in position to calculate the radial load and the thrust load on the supporting bearings due to the tooth pressure of bevel gears. In Fig. 5 AB represents the normal pressure on the gear tooth at the pitch line. We first resolve this into a component AC in a plane tangential to the pitch cone and a component CB in a direction perpendicular to this plane. The tooth pressure is regarded as a concentrated force at the center of load distribution, and both components pass through this point. AD represents the component perpendicular to the plane tangent to the pitch cone, both in magnitude and direction.

$$AD = CB = AC \tan 20° = T \tan 20°.$$

The latter may be resolved again into a component DE, parallel to the pinion axis (the thrust load on the pinion) and a component AE, which is in a plane perpendicular to the pinion axis.

$$DE = AD \sin \theta = T \tan 20° \sin \theta$$

$$AE = AD \cos \theta = T \tan 20° \cos \theta.$$

There are evidently two components of the tooth pressure in the plane perpendicular to the pinion axis, namely, AC and AE, and their resultant, AF, constitutes the radial load on the pinion bearing.

$$AF = \sqrt{AC^2 + AE^2} = \sqrt{T^2 + (T \tan 20° \cos \theta)^2}.$$

This equation applies to gears with 20° pressure angle.

The same formulae are applicable to the gear, but the results are different since θ, which occurs in each of them, has a different value. In the case of the differential gear it is not necessary to calculate the radial load on the pinion bearing (the pins) in this way, because these can be more simply calculated with sufficient accuracy from the rear axle torque, the mean distance of the pin bearing surfaces from the axis of rotation and the number of pins, as already explained. In calculating the thrust load on each pinion, account must be taken of the fact that teeth on opposite sides

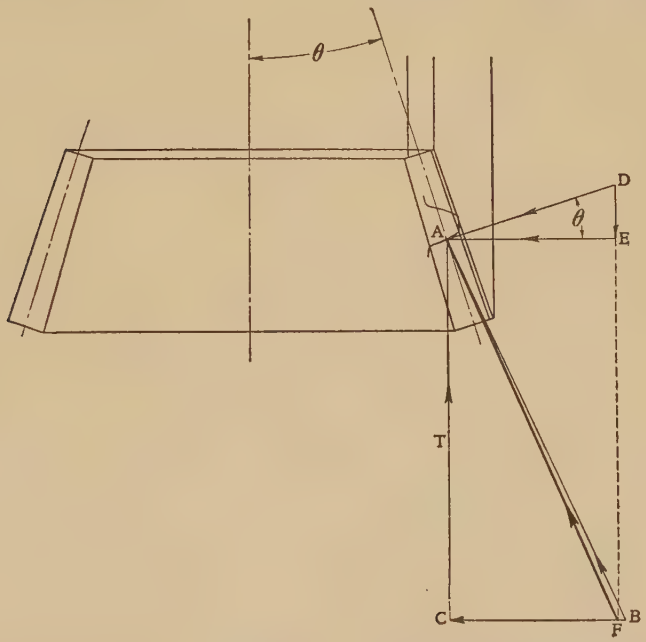


FIG. 5.—DIAGRAM OF COMPONENTS OF BEVEL-GEAR TOOTH PRESSURE.

are under pressure simultaneously. With a number of pinions equally spaced around the circumference there is no radial load on the side gears, and these therefore do not need large supporting surfaces.

Sample Calculation—In Fig. 6 is illustrated one of two four-pinion differentials standardized by the S. A. E., and we will apply some of the formula developed in the foregoing to it. The differentials have no official rating and we will see what the rating of the one illustrated should be for a

tooth stress of 100,000 lbs. per sq. in. as recommended in the foregoing. The pinions of this differential have 11 teeth and the side gears 20 teeth of 5–7 diametral pitch. The pitch radii evidently are

$$20/(2 \times 5) = 2 \text{ in.} \quad \text{and} \quad 11/(2 \times 5) = 1.1 \text{ in.},$$

hence the cone distance is

$$\sqrt{2^2 + 1.1^2} = 2.28 \text{ in.}$$

The face is 0.875 in., hence the value of the ratio a in the equation for tooth strength is

$$(2.28 - 0.875)/2.28 = 0.616$$

and

$$1 - a^3 = 0.766.$$

The value of y for a 11-tooth 5–7 pitch pinion is 0.095 and the circular pitch corresponding to 5–7 diametral pitch is 0.63. Allowing a working stress of 100,000 lbs. per sq. in., the equivalent tangential force on the pitch circle at the large end of the pinion is

$$F = \frac{100{,}000 \times 2.28 \times 0.63 \times 0.095}{3} \times 0.766 = 3480 \text{ lbs.}$$

This force acts on the rear axle shafts at a radius of 2 in., hence its moment is

$$2 \times 3480 = 6960 \text{ lb.-in.},$$

and since there are four pinions, each bearing on both sides, the capacity of the differential is

$$8 \times 6960 = 55{,}680 \text{ lb.-in. or } 4640 \text{ lb.-ft.}$$

The value of n in the equation for load distribution is

$$0.875/2.28 = 0.384,$$

and the value of m becomes

$$m = \sqrt{1 + \frac{0.384^2}{2}} - 0.384 = 0.83.$$

The mean pitch radius of the side gear therefore is

$$0.83 \times 2 = 1.66 \text{ in.}$$

Considering that there are four pinions and each meshes with a side gear on each side of it, the tooth pressure

$$T = \frac{55{,}680}{2 \times 4 \times 1.66} = 4190 \text{ lbs.}$$

The formula for thrust loads developed in the foregoing may be applied to advantage to the side gears, as the result will show not only the load on the thrust bearing surface of each of these gears, but also the force pressing the halves of the differential casing apart, which is of exactly the same magnitude. The value of tan 20° is 0.364 and the cone angle θ is

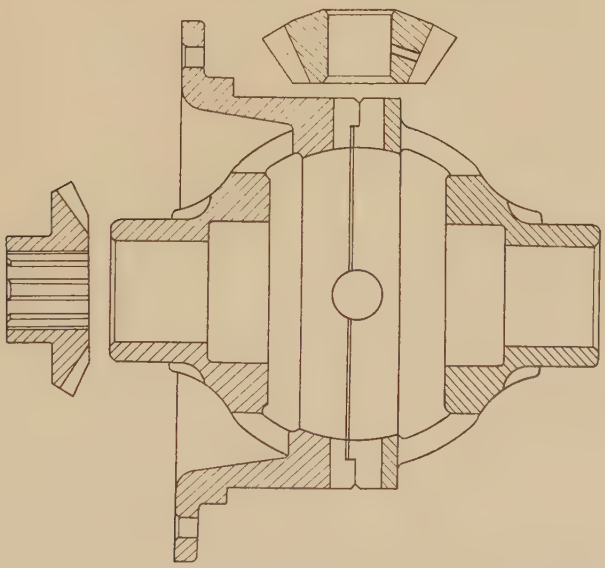


FIG. 6.—S. A. E.—STANDARD FOUR-PINION DIFFERENTIAL

61° 11′, whose sine is 0.877. Inserting these values in the equation for the thrust load we get

$$DE = 4190 \times 0.364 \times 0.877 = 1335 \text{ lbs.}$$

This same thrust is produced at each of the four side gear teeth in contact, and the total thrust therefore is

$$4 \times 1335 = 5340 \text{ lbs.}$$

This is also the force separating the halves of the casing.

While there are equal forces on the other side gear, these take the place of a reaction and do not add to the separating force.

Single-Piece-Housing Differential—The simplest design of differential gear is one having only two bevel pinions and a single-piece housing, of which an illustration is shown in Fig. 7. The inside of the housing is finished to a spherical surface having its center at the cone center of the gears. There are large openings in the sides of the housing through which the side gears can be inserted. The two pinions are then meshed with the side gears at the sides and the whole gear assembly is turned around the common axis of the side

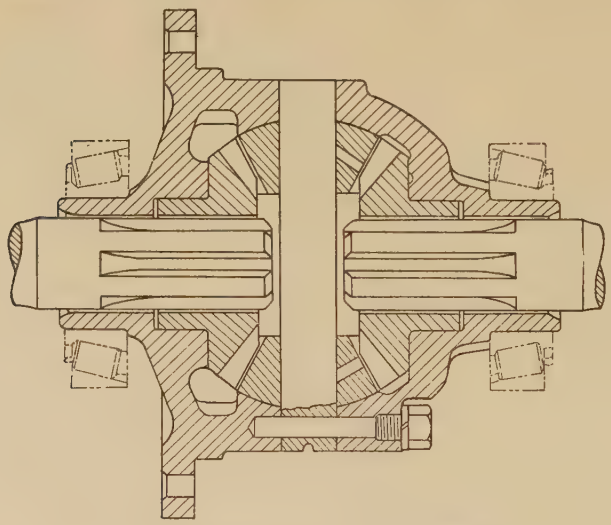


FIG. 7.—SINGLE-PIECE-HOUSING DIFFERENTIAL.

gears until the common axis of the pinions coincides with the axis of the holes for the pinion pin in the housing, whereupon the pin is introduced.

Constructional Details—In quantity production the housings are generally made of drop forgings. Single-piece housings cannot be drop forged and are made of malleable iron or steel castings, which materials are also used for two-piece housings when the production is too small to warrant the expense of drop forging dies. The housing—or one part of it—is formed with a flange to which the bevel crown gear is riveted. This flange is offset from the center plane of the

differential enough so that the axis of the bevel driving pinion lies in the center plane of the differential. In the case of worm-gear drive a flange is usually provided on each half of the housing and the worm wheel is clamped between the two flanges.

There are three methods of supporting the side gears, viz., in the housing, in the spider, and in both. The supporting bearings for the side gear need not be large, as there is no radial load due to the tooth pressure when there are two or more differential pinions uniformly spaced. There is, however, considerable axial thrust on the side gears, and ample thrust bearing surfaces must be provided.

The parts of the differential are lubricated by means of lubricant contained in the rear axle housing, the differential

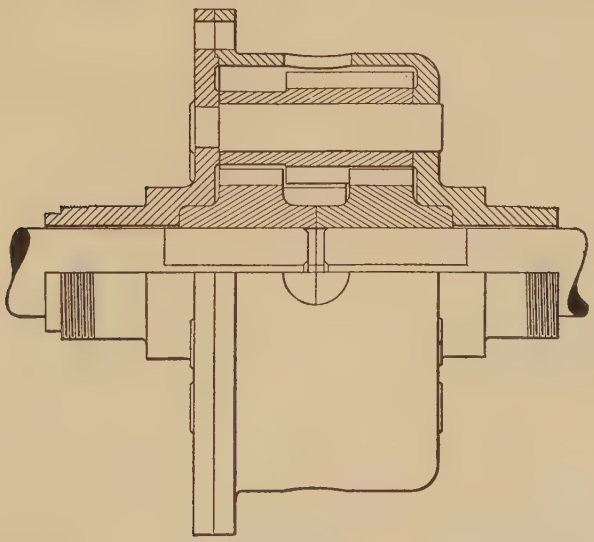


FIG. 8.—SPUR-PINION DIFFERENTIAL, LONGITUDINAL VIEW.

housing always being provided with openings so that this lubricant can get into it. In order that the lubricant may readily reach the bearing surfaces, a radial hole is usually drilled in each pinion, at the bottom of a tooth space, through to the bore, and oil grooves are cut in the thrust bearing surfaces of the side gears. In differentials for heavy-duty service bushings and washers are provided at all bearing surfaces so that by renewing these, any wear that may have occurred may be taken up. The hubs of the differential hous-

ing are turned to form seats for anti-friction bearings supporting the differential in the rear axle or jackshaft housing. The side gears are broached out to take the splined inner ends of the rear axle shafts, which latter clear the bore in the hubs of the differential housing.

The Spur Differential—Referring to Figs. 8 and 9 a spur differential consists of two spur gears mounted on the inner ends of the differential shafts, of a varying number of pairs of spur pinions and of a housing surrounding the whole. The spur pinions are of substantially double the width of the spur gears; the latter are placed some distance apart and the extra width of the pinions extends into this intermediate

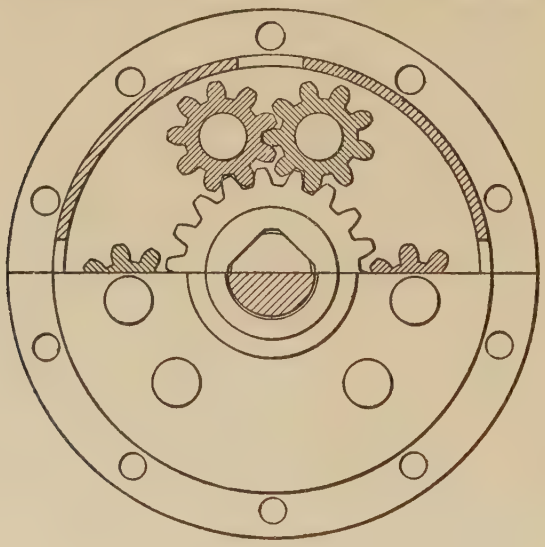


Fig. 9.—Spur-pinion Differential, Side View.

space where the two pinions of each pair mesh together. The action of this type of differential is exactly the same as that of a bevel differential.

The pinions of spur gear differentials are made with a small number of teeth, generally about ten, because any small increase in their size entails a large increase in the bulk of the differential housing. Stub teeth are preferably used, and some makers use a special form of mongrel teeth of still greater strength than stub teeth.

In calculating the parts for spur differentials the same tooth stresses can be figured with as in the case of bevel

differentials. The cases of spur differentials are made in two parts which are held together by bolts. The parts preferably should be provided with a telescoping joint, to insure the continued alignment of bearings. One part is usually made in the form of a circular plate and the other in the form of a cylinder open at one end, and the differential drive gear may be fastened to the housing by the same bolts which hold the two parts together.

Non-Slip Differentials—A large number of differentials have been invented which prevent a car from losing traction when one wheel stands on slippery ground. Most of them involve some form of one-way transmission device, that is, a mechanism by which power can be transmitted in one direc-

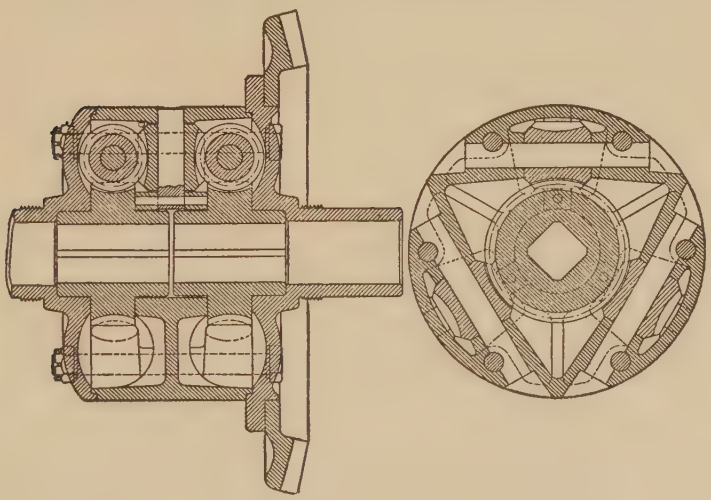


Fig. 10.—Non-slip Differential Gear (M. & S.).

tion but not in the other. With the ordinary differential, if one wheel is held from rotating and the frame or housing of the differential is rotated, the other wheel will be rotated at twice the speed of the housing. Also, if the housing is held from rotation and one wheel is rotated the other wheel will rotate in the opposite direction at the same speed. With one of the special differentials of the type referred to, if one wheel is locked or held from rotating, by turning the other wheel, the housing may be rotated, but it is impossible to turn the free road wheel by turning on the housing.

It is self-evident that such a differential does not equally divide the torque between the two driving wheels, for if it

did, then, when one wheel was spinning, the other wheel would have no more torque impressed upon it than the spinning one, which would be insufficient to propel the car. The relative torques impressed upon the two wheels respectively depend upon the resistance encountered by them. Ordinarily in straight-ahead motion, both wheels encounter substantially equal resistances, and the driving torque on both is therefore the same. But in turning a corner the outer wheel is compelled to run ahead of the differential housing or frame, and all the torque is taken by the inner wheel, the conditions then being the same as when one wheel has no traction.

One of the first differentials of the non-slipping type to appear on the market was the M. & S., illustrated in Fig. 10.

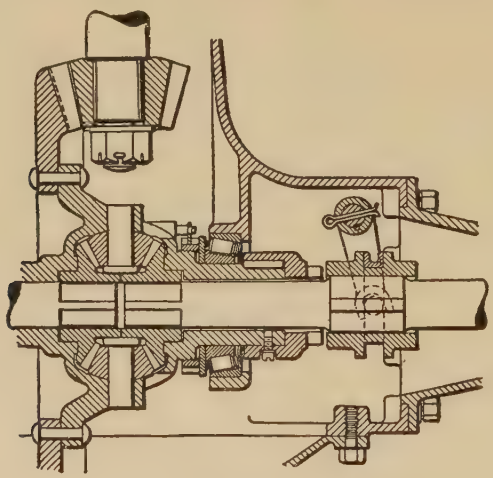


FIG. 11.—DIFFERENTIAL LOCK.

Each of the axle shafts carries a helical side gear and the differential housing carries three helical pinions with radial axes and six such pinions of which each one meshes both with one of the radial pinions and with one of the gears on the axle shafts. It is well known that in helical gears, if the helix angle of the driving gear is very small, power cannot be transmitted through the pair in the reverse direction, because the frictional resistance is too great, and this is the principle made use of in this differential.

Differential Lock—A few of the earlier trucks were provided with differential locks which enable the driver to put the differential gear out of operation at will. The problem

of working out a neat and all round satisfactory differential lock presents considerable difficulty, which is probably the reason that this device is not more extensively used. Fig. 11 illustrates a differential lock of typical design. A jaw clutch is provided, sliding on a squared section of one of the differential shafts, by which the differential housing may be locked to this shaft.

One fact that must be kept in mind by the designer when contemplating the use of either a non-slip differential or a differential lock is that with these the whole of the engine power may under certain conditions have to be transmitted by one axle shaft and one road wheel, and these parts may have to be made stronger than would otherwise be necessary. Of course, the torque load on these parts is still limited by the maximum ground adherence of the wheel.

CHAPTER VII

Bevel-Gear-Driven Rear Axles

Types of Passenger-Car Final Drive—In practically all of the earlier passenger cars, what is known as the final drive —that is, the drive from the spring-supported transmission to the rear axle or rear wheels—was either by chain or by bevel gears. The chain was soon discarded, for reasons which were outlined in Chapter I, and for a time the straight-bevel-gear drive practically monopolized the field. But in 1912 the spiral bevel gear was introduced, and it soon largely took the place of the straight bevel gear for automobile final drives, on account of its more nearly quiet operation. In some cases worm gear drive is used for passenger cars, especially in England, where worm and wheel drive for motor vehicles originated. Another type of gearing for automobile final drives, known as the hypoid gear, was introduced in 1926. It is in some respects intermediate between the spiral bevel and the worm gear, in that the relative amount of sliding motion in it is greater than in the spiral bevel, but less than in the worm gear. The specific object aimed at in the introduction of the hypoid gear was a lowering of the propeller shaft, which permitted of lowering the floor of the body, thus making possible a generally lower construction and greater safety at high speeds. Still another intermediary type, more closely approaching the worm gear, has been under development for some years but has not yet been applied in production. The driving member is a worm, but the teeth of the driven member, instead of being on the circumference of the rim, are on the side thereof.

Types of Rear Axle—Rear axles are divided into live axles and dead axles, though these terms are not much used now, since an overwhelming majority of all axles are of the former type. A live axle is an axle through which the propelling power is transmitted to the driving road wheels, and which therefore comprises rotating members, while a dead axle is one which merely carries a fraction of the weight of the frame

and body. Usually the weight-carrying and propelling or driving members of live axles are separate, the former being of tubular or hollow form and the driving members located inside this tubular carrying member. There is another type of axle, however, in which there is a final gear reduction directely at the driving wheels, in which the driving members are located adjacent to the carrying member, which may be of either solid or tubular section.

Parts of Live Axle—The part of a live axle which supports the load is known as the housing, while the parts which transmit the driving torque to the wheels are known as axle shafts. The axle housing comprises a central enlarged portion, known as the center housing or driving gear housing, enclosing the bevel driving gears and the differential gear; and lateral extensions thereof which, if they are separate pieces, are known as axle tubes. The driving gear housing also has a forward extension in which are mounted the bearings supporting the bevel pinion shaft. To the outer ends of the axle tubes are secured the brake-supporting discs or spiders, and in some cases housings for the axle bearings, while intermediate between the driving gear housing and the brake spiders, spring seats and brake shaft-supporting brackets may be either formed on the axle housing or secured thereto in a suitable way.

Live axles are subjected to three principal stresses, namely, torsional stresses due to the driving torque and the braking torque; bending stresses due to the weight resting on the axle and to the driving force, and bending stresses resulting from transverse loads on the wheels due to centrifugal force when turning corners at speed, to hitting obstructions while skidding, etc.

Classification of Live Axles—A live axle in which the axle shafts are subjected to torsional stresses only is known as a full-floating axle. An axle in which the shafts are subjected to all three of the stresses enumerated above is known as a semi-floating axle, while one in which they are subjected only to torsional stresses and to bending stresses due to transverse loads on the wheels but not to bending stresses due to the load on the axle and the propelling force, is known as a three-quarter-floating axle. Full-floating, semi-floating and three-quarter floating axles are shown in Figs. 1, 2, and 3, respectively. The center portions of all three types are exactly alike. In each the differential gear is supported in bearings which in the drawings are shown mounted directly in the driving gear housing, but which may also be mounted in a so-called differential carrier which is set into the hous-

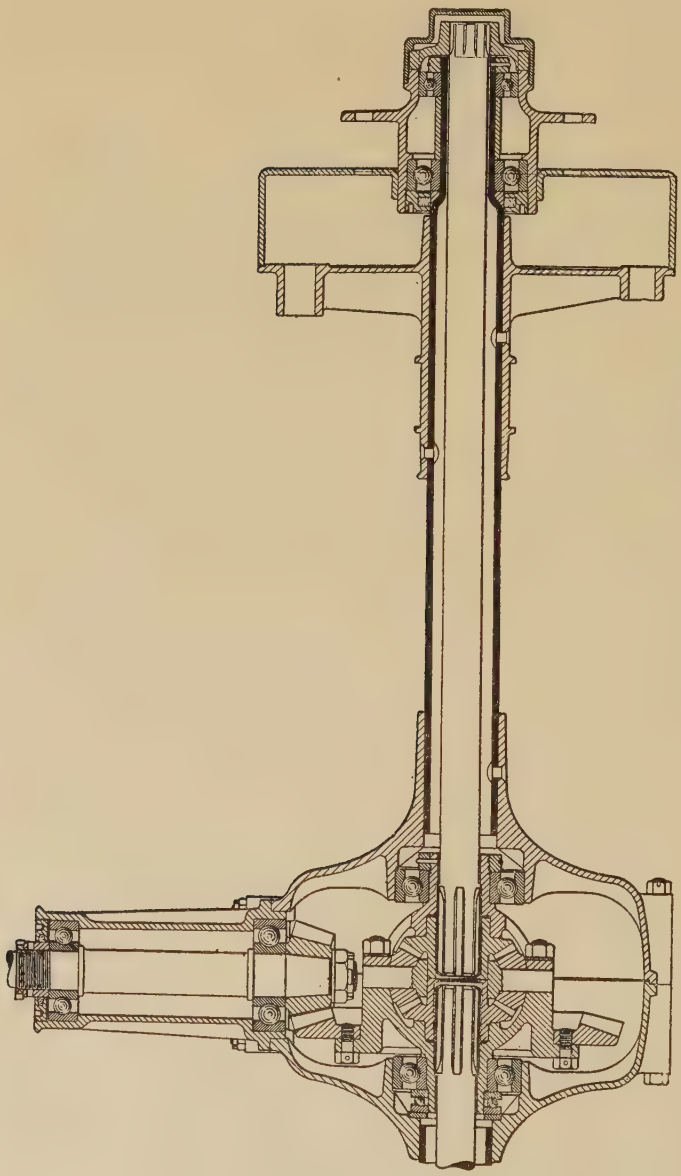


FIG. 1.—FULL-FLOATING TYPE OF REAR AXLE.

ing. In the full-floating type of axle each axle tube is reduced in diameter near its outer end and carries two bearings on which the wheel hub revolves. The axle shaft protrudes from the tube a short distance and is placed in driving relation with the hub, through a driving dog or clutch. In the drawings, which are more or less diagrammatic, the axle shafts are shown to have splined joints with the differential side gears and the driving clutches. The driving clutch is connected to the wheel hub by means of intermeshing teeth or by bolts.

In the three-quarter-floating axle there is only a single bearing on the outer end of the axle tube, which is located centrally with respect to the rear wheel. The outer flange of the rear wheel hub is securely fastened to the axle shaft, usually by keying on a tapered seat. Thus the road wheel forms a rigid unit with the axle shaft, and any blow which the wheel may receive in skidding is transmitted to the shaft.

In the semi-floating axle the axle tube does not extend into the wheel hub, hence the load on the axle tube or housing can be transmitted to the wheels only through the intermediary of the axle shafts. The bearings at the ends of the axle are mounted, not on the axle tube, but in a bearing housing which is formed on or secured to the end of this tube. The wheel hub is securely fastened to the protruding portion of the axle shaft. It is immediately apparent that in this case the axle shafts are subjected to all three of the stresses previously referred to.

One advantage of the full-floating type of axle over the other types is that in the event of the breakage of an axle shaft the axle is not let down, and the car can be towed without difficulty. Breakage of the axle shaft, if occurring at high speed, evidently does not involve the risks to the passengers that it does in the cases of the other axle types. In fact, so far as axle shaft stresses are concerned, the full-floating type of axle is the best, its shafts being subjected to the least stresses; it is followed by the three-quarter floating and the semi-floating types in that order. From the standpoint of bearing loads, however, the semi-floating type of axle has the advantage in some respects. We will compare a full-floating and a semi-floating axle with regard to the bearing loads due to lateral shocks on the wheels.

Bearing Loads Due to Lateral Forces—Fig. 4 is a diagram of a full-floating type of axle in which the wheel is shown to be subjected to a lateral force F at the rim. Let R be the radius of the wheel; D, the distance between centers of the wheel bearings; L_1 and L_2, the radial reactions of the

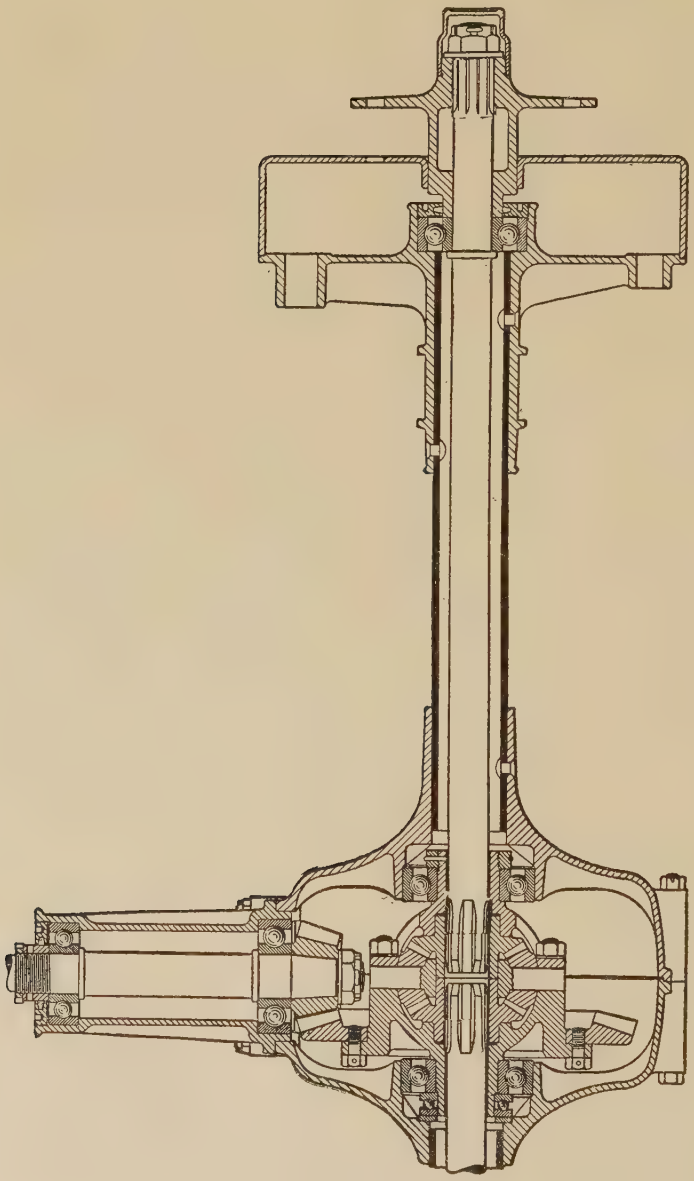


FIG. 3.—SEMI-FLOATING TYPE OF REAR AXLE.

wheel bearings on the wheel hub, and T, the thrust reactions of the two bearings (supposing these to take the thrust load). Then, since the moments around any axis must vanish, we have

$$FR = L_1D \quad \text{and} \quad FR = L_2D$$

$$L_1 = \frac{R}{D}F = L_2.$$

In practice the expression R/D has a value of about 4, hence, when the wheel receives a lateral shock, each of the two wheel bearings is subjected to a shock load about four times as great as the shock on the wheel. In the case of the inner bearing this adds to the normal load, whereas in the case of the outer bearing the shock load and the normal load are opposite in direction and the resultant load on the bearing is the difference between the two. Incidentally it may be pointed out that the thrust load on the bearing is equal to the shock, since $T = F$.

The semi-floating axle is shown in diagram in Fig. 5. In this case

$$L_1D = FR = L_2D \quad \text{and} \quad L_1 = \frac{R}{D}F = L_2.$$

In practice $R/D = 0.8$ approximately, so that L_1 and L_2 are each equal to only about 80 per cent of the shock F. We found that in the case of the full-floating axle they are equal to about four times the shock, hence the bearing loads due to lateral shocks on the wheel are roughly five times as great in the full-floating type as in the semi-floating and three-quarter-floating types of axle.

Axle Shafts—The shafts of driving axles are always made of alloy steel of high physical properties, and the factor of safety is kept as low as permissible. The reason for this is that any increase in the diameter of the axle shaft necessitates a corresponding increase in the diameter of the axle tubes, bearings, etc., and entails a comparatively large increase in weight, and that dead or unsprung weight. In the case of full-floating axles the shafts are subjected to torsional stresses only and are calculated by means of the equation for torsional strength given in the Chapter on Universal Joints and Propeller Shafts.

Axle Shaft Steels—The steel most frequently used for rear axle shafts is S. A. E. No. 3140, a chrome-nickel steel containing 1–1.50 per cent Ni, 0.45–0.75 per cent Cr, 0.50–0.80 per cent Mn and 0.35–0.45 per cent C. This steel, when nor-

malized at 1600°–1700° F., quenched in oil at 1475°–1525° F. and drawn at 800° F., shows a tensile strength of 175,000 lbs. per sq. in. and an elastic limit of 150,000 lbs. per sq. in. Another steel sometimes used is the S. A. E. No. 3240, a chrome-nickel steel containing 1.50–2.00 per cent Ni, 0.90–1.25 per cent Cr, 0.30–0.60 per cent Mn and 0.35–0.45 per cent C. This steel when normalized at 1625°–1725° F., quenched in oil at 1475°–1525° F., and drawn at 800° F., shows a tensile strength of 200,000 lbs. per sq. in. and an elastic limit of 180,000 lbs. per sq. in. Nickel steel of the 3½ per cent grade (No. 2335) as well as chromium steel and molybdenum steel are also used for these shafts.

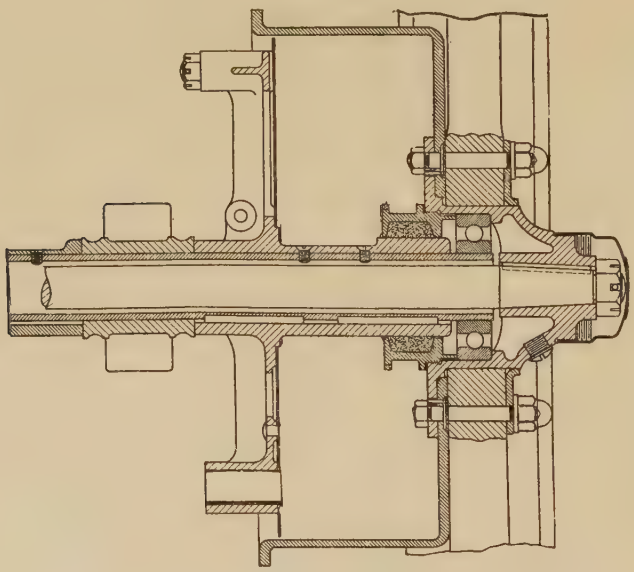


Fig. 3.—End of Three-Quarter-Floating Rear Axle.

Ordinarily the shafts are drawn at such a temperature that they will be sufficiently soft to permit of machining after heat treatment, the heat treatment being applied at the steel mill and the machining done at the car or axle plant. This makes it necessary to keep the Brinell hardness down to about 300, and, of course, also limits the yield point and the ultimate strength of the steel. If greater strength is wanted the shafts are machined before heat treatment and only certain operations calling for great accuracy, such as cutting of keyways and splines, is left until afterwards. The Brinell

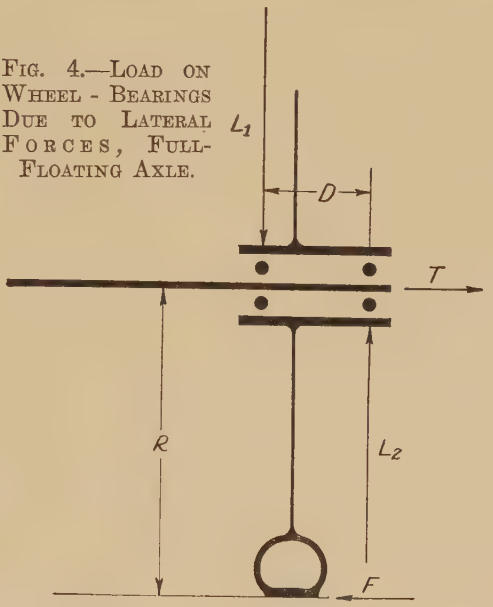


FIG. 4.—LOAD ON WHEEL - BEARINGS DUE TO LATERAL FORCES, FULL-FLOATING AXLE.

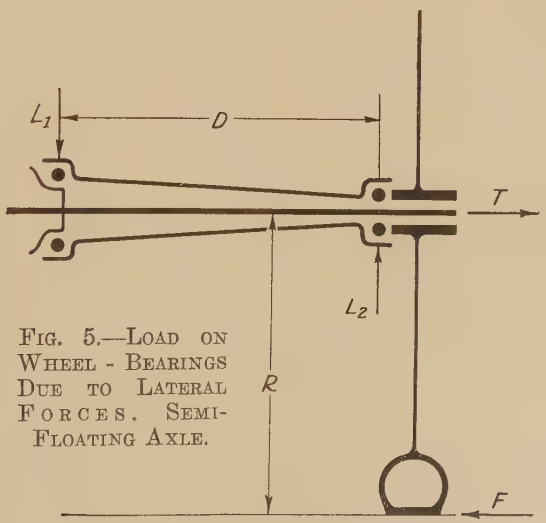


FIG. 5.—LOAD ON WHEEL - BEARINGS DUE TO LATERAL FORCES. SEMI-FLOATING AXLE.

hardness can then be carried up to 350 or 375. For the greatest strength the shafts are fully machined in advance and drawn at a low temperature to give a Brinell hardness of close to 400. This calls for cold straightening after heat treatment.

Calculation for Torsional Strength—A factor of safety of 2 can be used with the torsional strength formula by making use of the elastic limit of the material in torsion, or a factor of safety of 2.75 by making use of the apparent ultimate strength in torsion. Torsional strengths of various axle steels were given in Chapter V. The joints of the axle shafts with the differential and the driving clutch are usually of the splined type, and for heavy vehicles the ten-spline joint is preferably used, as the depth of the splines is less, and the splines add nothing to the torsional strength. The ends of the shafts are usually upset and the splines cut in these upset portions. At the inner ends of the splines a rounded shoulder is cut, and the spline grooves or flutes also run out gradually, so there can be no concentration of stress at any point.

Axle Torque Calculation—In calculating the maximum torque which the axle shafts will be called upon to sustain we may either start with the maximum engine torque, multiply that by the greatest reduction in the gearbox and by the rear axle reduction ratio, and then divide by two (because the conventional differential always divides the rear axle torque equally between the two shafts); or we may start with the maximum permissible weight on each rear wheel and multiply that by a friction coefficient of 0.6, which is nearly the maximum encountered in practice. This will be the maximum tangential force on the driving wheel, and when multiplied by the radius of the wheel, gives the maximum torque on the axle shaft. If a self-locking differential or a differential lock is used, account should be taken of the fact that more than half of the rear axle torque which the engine can produce through the gearing used, can come on one of the shafts, if its wheel has sufficient ground adherence. The axle shafts also have to transmit the braking torque if a transmission brake is used, but this also is limited by the ground adherence of the wheels. With skid chains a coefficient of ground adherence of more than 0.6 might be obtained, but this would be taken care of by the factor of safety.

Shafts of Semi-Floating Axles—In calculating the necessary dimensions of the shafts of semi-floating axles both the torsional load and the bending load due to the weight on the axle must be taken into account. The point of support

for the axle shaft is at the center of the road wheel, and the load is transferred to the axle housing through the bearing, hence a bending moment on the axle shaft is created whose moment arm (l Fig. 6) is equal to the distance between the road wheel center plane and the center of the outboard bearing. The wheel load, of course, is not accurately known in advance but will have to be judged from that of an existing similar vehicle.

The maximum bending moment on the shaft occurs at the center of the bearing and is equal to wl, where w is the load carried by one of the wheels, and l the distance between the center plane of the wheel and the center of the bearing. Let T equal the maximum torque on one of the axle shafts in pound-feet; that is, one-half the product of the maximum engine torque by the low gear reduction ratio and the bevel gear reduction ratio. If the diameter of the shaft is d,

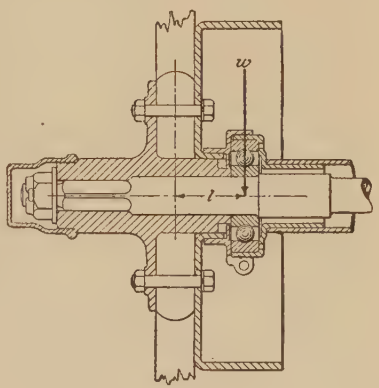


FIG. 6.

the distance c of the outermost fibre from the neutral axis is $d/2$ and the moment of inertia I of the cross-section is $\pi d^4/64$, hence, inserting in the well-known formula for bending stress

$$S_b = \frac{Mc}{I},$$

we have

$$S_b = \frac{32wl}{\pi d^3}.$$

The polar moment of inertia of the circular section is $\pi d^4/32$ and inserting in the formula for torsional stress

$$S_t = \frac{Mc}{J},$$

we get

$$S_t = \frac{192T}{\pi d^3}.$$

These two stresses can be combined by means of the equation

$$S_c = \tfrac{1}{2}S_b + \sqrt{S_t^2 + \tfrac{1}{4}S_b^2},$$

the derivation of which can be found in text-books on Mechanics. Inserting the values found for S_b and S_t, we get:

$$S_c = \frac{16wl}{\pi d^3} + \sqrt{\left(\frac{192T}{\pi d^3}\right)^2 + \tfrac{1}{4}\left(\frac{32wl}{\pi d^3}\right)^2}$$

$$= \frac{1}{d^3}\left[\frac{16wl}{\pi} + \sqrt{\left(\frac{192T}{\pi}\right)^2 + \tfrac{1}{4}\left(\frac{32wl}{\pi}\right)^2}\right].$$

Hence

$$d = 1.72\sqrt[3]{\frac{wl + \sqrt[2]{144\,T^2 + w^2l^2}}{S}}.$$

This diameter is required at the outboard bearing. The bending moment decreases uniformly from the center of the bearing to the center of the road wheel and the center of the differential side gear hub, respectively, and if the lightest possible construction is desired the shaft diameter may be decreased from the value calculated by the above equation as required at the outboard bearing, to the diameter required for the torsional stresses only at the center of the road wheel and the master gear, respectively. The bending moment, of course, does not vanish at the differential, but its effect on the shaft stresses there is negligible.

In choosing the value for S in the above equation, a rather high factor of safety must be allowed—much higher than that used in calculating the necessary section of portions of the shaft subjected to torsional stresses only. One reason for this is that severe road shocks will increase the bending stresses beyond the normal to a greater extent than severe clutch action can increase the torsional stresses; besides, the outer end of the shaft of a semi-floating axle is subjected to repetition reversing loads, which call for an extra high factor of safety in order to obviate fatigue failure. A factor of safety of 7, based on the ultimate strength in tension, gives good results. When the rear wheels are fitted with dual pneumatic tires the possibility of the whole of the weight being carried by the outer wheel alone should be taken into account, and the distance l should be measured from the center plane of the outer tire to the center plane of the outboard bearing.

Where the shaft has a splined joint with the differential side gear the diameter at the bottom of the splines is made slightly greater than the calculated diameter, which is used just inside the splined section. Otherwise the splined section is likely to fail through fatigue. Such a failure is characterized by a fracture showing radial lines from the axis to the

bottom edges of the splines. The splined upset portion joins the smaller-diameter, solid section with a large fillet. Any local weakening of the shafts, such as might result from stenciling thereon, should be carefully avoided, as this often leads to fatigue failure, in which a minute initial crack or fissure gradually widens until the shaft breaks.

Types of Axle Housing—Two general types of axle housing are used for bevel gear-drive axles, viz., the built-up type with cast center housing and inserted drawn tubes, and the pressed steel housing. Where bevel-gear drive is used for very heavy units, the whole axle housing may be of cast steel or malleable iron. The built-up type of axle, which is not very extensively used any more, is illustrated in Figs. 1 and 2. The center housing is made of malleable cast iron and is split centrally in a vertical plane. Each half is formed with a hub into which an axle tube of seamless drawn steel tubing is fitted by riveting. The differential and bevel driven gear evidently must be put in place before the halves of the housing are bolted together. One objection to the construction is that it does not permit of examining the differential and driving gears and their bearings without taking the axle apart. There is a somewhat different form of built-up axle which is free from this objection. In this the central housing is made in a single casting with a large opening at the rear through which the differential assembly can be introduced.

Rear axle housings of the so-called double banjo type, of pressed steel, are widely used in passenger car practice. The housing is formed in halves in the press, which are later welded together either electrically or by the acetylene torch. The weld is generally at the top and bottom, and while it is mostly a seam weld, in large quantity production the halves may be joined together by the butt-welding process. Large circular openings are formed in the central housing at both front and rear, and the edge of the housing at these openings is reinforced by turning the metal over in the press or by providing reinforcing rings of flat material on the inside. This is necessary in order to get sufficient thread for the bolts which hold the differential carrier in place. This reinforcing flange, of course, also adds to the stiffness of the housing in the vertical plane.

Details of Pressed Steel Housings—Pressed steel axle housings are manufactured in a number of different designs. Only the housings for axles of passenger cars of the lighter type are made of pressings alone, and axles for heavier cars are generally stiffened by tubular members. A plain pressed

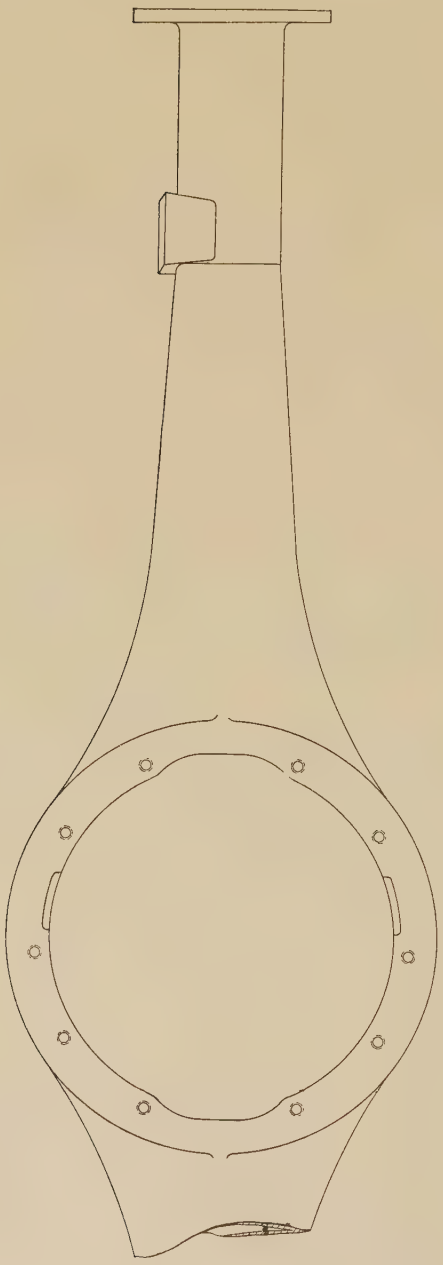


FIG. 7.—AXLE HOUSING ENTIRELY OF PRESSED STEEL.

steel housing is illustrated in Fig. 7. The central part of the housing is in the form of a shallow cylindrical shell with inward flanges on both sides, which in this case is reinforced by an inner shell of pressed steel which fits snugly in it. The tubular portions of the axle housing enlarge slightly in the horizontal and more pronouncedly in the vertical direction as they approach the shell. The flange at the forward side of the shell is cut away at certain points to permit the introduction of the rear axle crown gear and the differential bearings with a minimum outside diameter of the shell and therefore maximum ground clearance. The openings in the inner

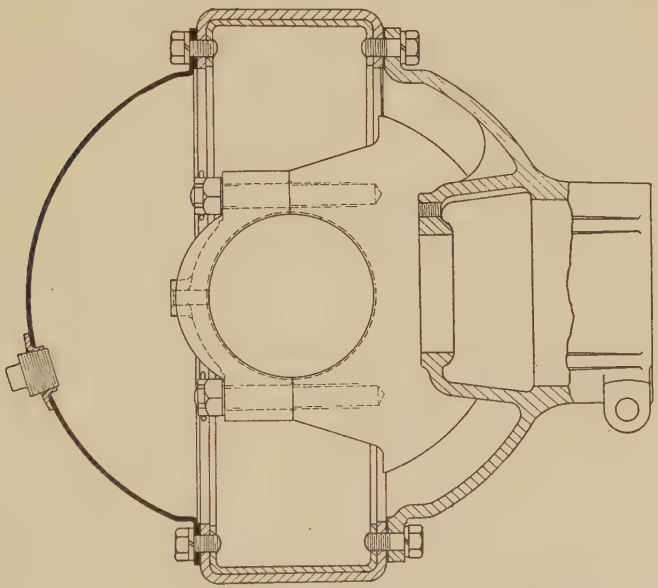


FIG. 8.—CENTRAL TRANSVERSE SECTION THROUGH PRESSED STEEL AXLE HOUSING AND DIFFERENTIAL CARRIER.

shell through which the axle shafts pass are not much larger than the upset ends of the shafts, the wall serving as an oil dam. Figure 8 is a cross-section through the center housing at right angles to the axis of the axle.

Fig. 9 shows a pressed steel rear axle housing with a reinforcing tube of seamless steel forced into it. This comes up close to the shell-shaped central section, where it is supported in the pressed steel housing by a pressing of flanged annular shape. At the outer end the tubular insert comes

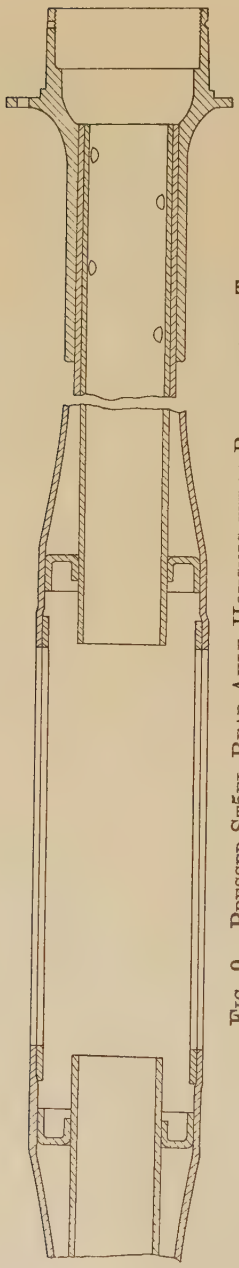


FIG. 9.—PRESSED-STEEL REAR-AXLE HOUSING WITH RE-INFORCING TUBE.

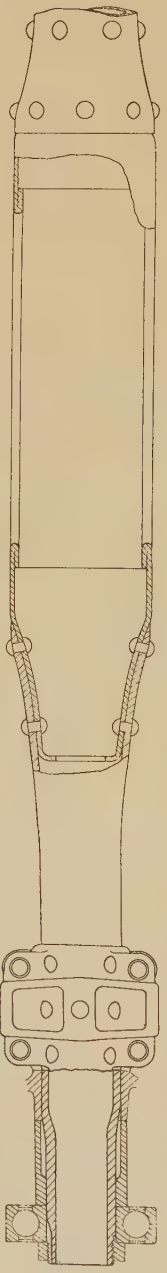


FIG. 10.—REAR-AXLE HOUSING WITH PRESSED-STEEL CENTER AND TUBULAR END SECTIONS.

flush with the end of the pressed steel housing. In another design the tubular insert is made of an inside diameter equal to the inside diameter of the pressed steel housing at the outer end, and extends only part way from the central section to the end of the housing, its outer end being tapered to fit accurately in the tapered pressed steel housing.

Still another design of pressed steel housing is shown in Fig. 10. Here only the central section is of pressed steel, the outer sections being rolled flared tubes in which the wall thickness is varied in accordance with the varying loads along the length of the housing. This construction is of particular advantage for high-powered cars in which it is desired to keep down the weight. The soft steel (of about 0.20 per cent carbon) of which the pressings of the axle housing are made has an elastic limit of only about 40,000 lbs. per sq. in. and an ultimate strength of 70,000 to 75,000 lbs. per sq. in., while the tensile strength of heat-treated nickel steel tubing is more than twice and the elastic limit nearly twice as great. Thus considerable weight can be saved by the use of alloy steel tubing for that part of the axle housing which is tubular in form. In the axle housing shown in Fig. 10 the pressed steel and tubular portions are joined by riveting.

To the outer end of the axle housing are secured the brake carrier, the spring seat and—in the case of semi-floating axles—the bearing housing. These parts may all be in a single casting (usually of malleable iron), or they may be separate castings and forgings, and they may be secured to the housing by riveting or welding, riveting being always employed in the case of built-up axles.

Stresses on Axle Tubes—In discussing the stresses on the housing of a driving axle we have to distinguish between three different cases, viz.,

(1) When the car is being started;
(2) When the car is being driven;
(3) When the car is being braked.

The first two cases are similar in that the stresses produced are the same in kind and differ only in magnitude. Since the clutch is capable of transmitting more torque than the engine is capable of generating, if the clutch should be allowed to engage suddenly the flywheel would supply this extra torque and would subject the axle housing (as well as the axle shafts) to extra stresses. The limiting stress which can be thus produced depends upon the clutch capacity or adjustment. When the car is being started or driven forward, the loads and reactions on the axle structure are as

shown in Fig. 11. There is a downward pressure w on each of the two spring seats. Then there is the weight w_1 of the axle itself, which may be considered concentrated at the center. Then there are also the loads b b on the differential bearings, the load b' on the pinion bearing and the reaction c at the point of support of the torque arm on the frame. Finally there is the reaction P of the road surface on the driving wheel causing the forward motion of the car.

It can easily be shown that the bearing loads b b are equal to b', hence the pressures on the inner bearings create no stress in the axle tubes. Also, the moments of forces b' and c around the rear axle axis are equal and opposite, and therefore have no effect on the axle tubes. There remain only the vertical bending moments due to the weights w w on the spring seats, the vertical bending moment due to the weight w_1 of the axle, and the horizontal bending moment due to the propelling thrust P.

When the brakes are applied we have the same moments in the vertical plane due to the weight resting on the spring seats and the weight of the axle. The horizontal moment is in the opposite direction and is proportional to the maximum braking force instead of to the maximum propelling force, which former is at least equal to the latter. In addition we have in this case a torsion moment on the tubes, since the brake supports are secured to the outer ends of the axle tubes, and when the brakes are applied the friction between brake band and drum tends to carry the brake supports around with the drums. This tendency is counteracted by the torque tube or rod which is generally fixed to the axle housing near its middle.

It will thus be seen that when braking, the axle tubes are subjected to the same bending stresses as when driving, and, besides, they are subjected to torsion. Hence the combined stress is greatest when the brakes are applied, and only this case needs to be considered. Owing to the fact that the weight of the axle is not known in advance, and since it is small in comparison with the weight on the springs, it is advisable to neglect this factor.

In respect to the vertical load, the axle housing forms a beam freely supported at both ends and loaded at the center and two intermediate points, the equivalent point of support being at the center of the road wheel in the case of full floating and three-quarter floating axles and at the center of the outboard bearing in the cases of a semi-floating axle. The bending moment increases from nothing at the point of support to the maximum at the center of the spring seats and is constant between spring seats. (See Fig. 12.)

In respect to the horizontal bending moment, each half of the axle may be considered as a cantilever, the middle of the axle being the fixed end of the lever and the load being applied to the outer, free end. The variation of all of the moments is shown in the three-plane diagram, Fig. 12.

The case is more favorable, of course, if the driving thrust is transmitted from the rear axle housing to the chassis frame through the chassis springs or radius rods instead of through

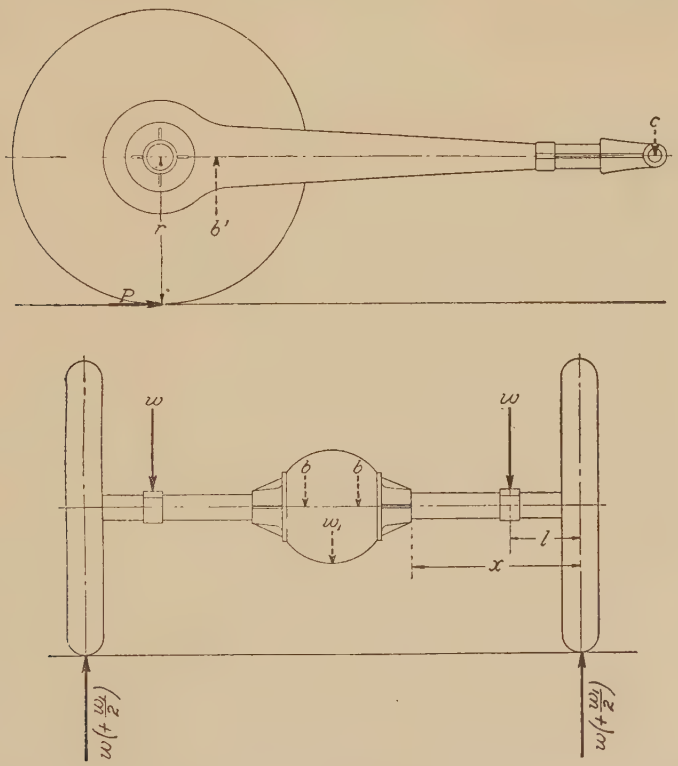


Fig. 11.—Loads on Housing When Car is Being Driven Forward.

a torque tube at the center, as in that case the arm through which retarding force or braking force acts is equal to the distance between the wheel center or outboard bearing center and the center of the spring seat. The brake force moment diagram then assumes substantially the same form as the wheel reaction moment diagram A in Fig. 12.

Now, let us take any section of the tube inside the spring seat at a distance x from the center plane of the road wheel. The moment at this section due to the weight on the spring seat is wl, and the moment due to the retarding force of the brakes is Px. The maximum retarding force P is reached when the wheels are locked on dry macadam surface, under which condition the coefficient of friction with rubber tired wheels is about 0.6. Hence, $P = 0.6w$ and $Px = 0.6wx$. These two bending moments are in planes at right angles to

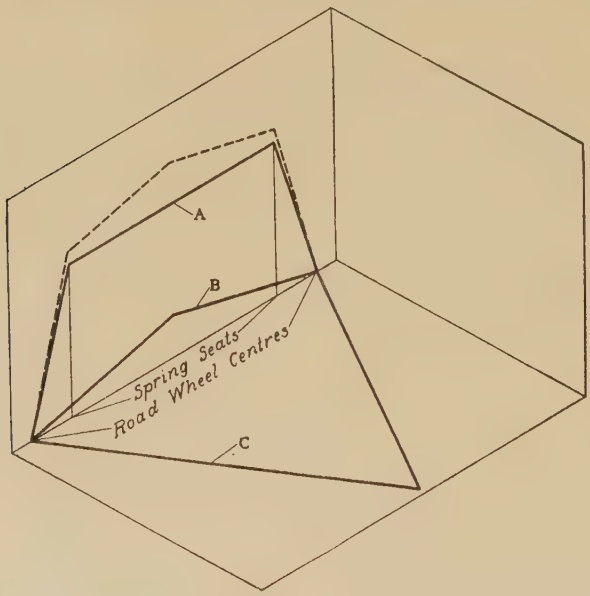


FIG. 12.—BENDING MOMENTS ON AXLE HOUSING.

each other and their resultant (Fig. 13) is equal to the square root of the sum of their squares. That is,

$$M_b = \sqrt{w^2l^2 + 0.6^2w^2x^2}$$

$$= w\sqrt{l^2 + 0.36\,x^2}.$$

Denoting the radius of the road wheel by r, the torsional moment on the axle tube is

$$M_t = Pr = 0.6\,wr.$$

The unit bending stress is

$$S_b = \frac{M_b c}{I},$$

and the unit torsional stress,

$$S_t = \frac{M_t c}{J}.$$

The combined stress is

$$S_c = \frac{M_b c}{2I} + \sqrt{\left(\frac{M_t c}{J}\right)^2 + \frac{1}{4}\left(\frac{M_b c}{I}\right)^2}$$

Remembering that for a circle $J = 2I$, we may write this in the form

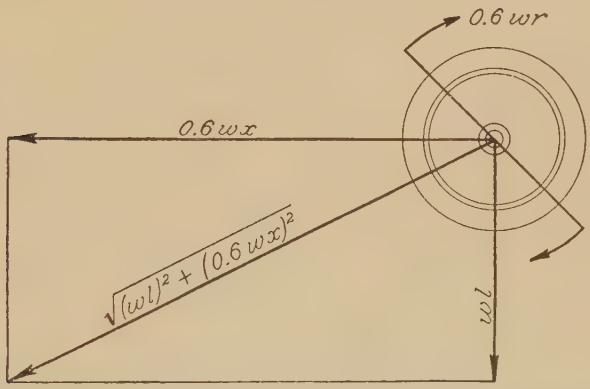


FIG. 13.—COMPOSITION OF MOMENTS AND COUPLE ON AXLE HOUSING.

$$S_c = \frac{c}{2I}(M_b + \sqrt{M_t^2 + M_b^2}).$$

Now let D denote the outside diameter of the axle tube and d the inside diameter. Then

$$c = \frac{D}{2}$$

and

$$I = \frac{\pi(D^4 - d^4)}{64}.$$

Inserting these values and those of M_t and M_b found above in the equation, for S_c we get

$$S_c = \frac{\frac{D}{2}}{\frac{2\pi(D^4 - d^4)}{64}} (w\sqrt{l^2 + 0.36\,x^2} + w\sqrt{0.36\,r^2 + l^2 + 0.36x^2})$$

$$= \frac{5Dw}{D^4 - d^4} (\sqrt{l^2 + 0.36\,x^2} + \sqrt{l^2 + 0.36\,x^2 + 0.36\,r^2})$$

Denoting the part in parentheses by y we may write

$$(D^4 - d^4)S_c = 5Dwy$$

$$D^4 S_c - 5Dwy = d^4 S_c$$

$$d^4 = D^4 - \frac{5Dwy}{S_c}$$

$$d = \sqrt[4]{D^4 - \frac{5Dwy}{S_c}},$$

and replacing the value of y:

$$d = \sqrt[4]{D^4 - \frac{5Dw\,(\sqrt{l^2 + 0.36x^2} + \sqrt{l^2 + 0.36x^2 + 0.36\,r^2})}{S_c}}$$

Permissible Stresses in Axle Tubes—In calculating the dimensions of the axle housing a factor of safety of 7, based on the ultimate strength of the material, can be figured with. This may seem very high in view of the fact that a factor of safety of only 2.75 is allowed for the axle shafts, but in the case of the housing it is a question not only of preventing failure but of keeping down the deflection of the housing under load. (In the case of semi-floating axles this applies also to the shafts.) An axle housing of insufficient rigidity bends and causes the road wheels to spread at the bottom, which is very unsightly and also impairs the operating conditions of the bearings. It would therefore really be most logical to calculate the rear axle on the basis of the deflection caused in it by the load, but this is difficult because of the change in the section of the housing.

The stress in pressed soft steel should be limited to about 10,000 lbs. per sq. in.; in medium carbon steel drawn tubing to 15,000 lbs. per sq. in., and in alloy steel tubing, heat treated, to 20,000–25,000 lbs. per sq. in. These stresses, of course, would make the pressed steel housing the most rigid, as rigidity depends upon the coefficient of elasticity, which is about the same for all steels.

Differential Carrier—As already pointed out, practically all American designs of bevel gear rear axles embody the feature of a differential carrier. A differential carrier is a casting that is secured to the axle housing by bolts and contains bearings for the bevel pinion shaft and the differential. One advantage of this construction is that the bearings of

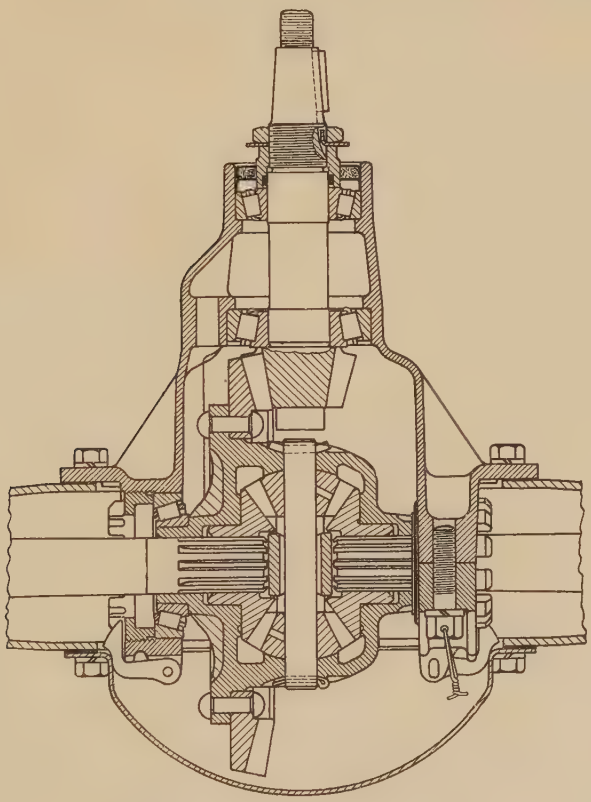


FIG. 14.—DIFFERENTIAL CARRIER.

both the driving and the driven member of the bevel-gear set are in the same piece and therefore cannot so readily get out of alignment. Another is that the whole driving gear and differential assembly can be removed from the axle without removing the latter from the vehicle.

A typical differential carrier, assembled in the axle, is shown in Fig. 14. The bevel gear pinion shaft is supported

on two taper roller bearings, which not only take radial loads but also thrust loads in both directions. In order to keep down the load on the bearing adjacent to the pinion, the two bearings are spaced considerably. No means of adjusting the bevel gear pinion axially are provided, though such adjustment could be made by placing a washer between the pinion and the bearing behind it. The differential also is sup-

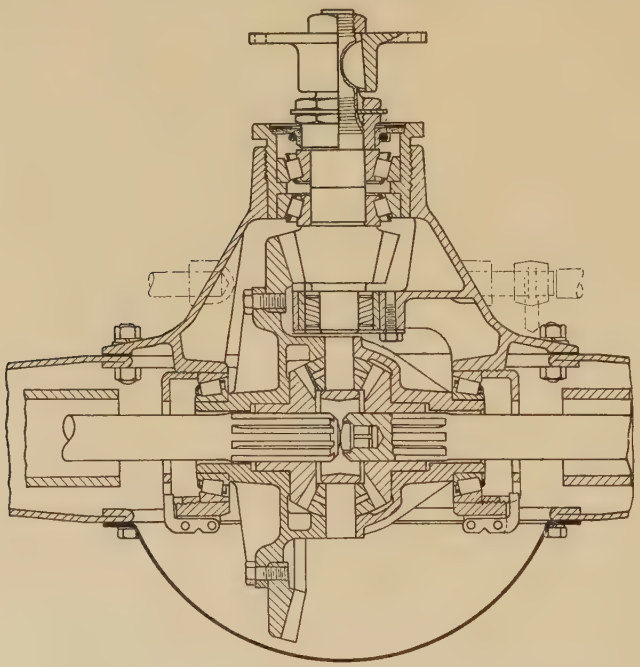


FIG. 15.—STRADDLE-MOUNTING FOR BEVEL PINION.

ported in two taper roller bearings, and in this case means of adjustment are provided. The outer race of the bearing seats in a smooth bore in the differential carrier and is backed up by a castellated annular nut. The differential-bearing hubs are split, the caps being held to the differential carrier proper by two cap screws each. These cap screws are secured by lock washers and the castellated differential adjusting nuts are locked by dogs which in turn are secured by the inspection hole cover or rear cover.

Axle Lubrication—Lubrication of rear axles is effected by partly filling the housing with a heavy oil. This oil often

has difficulty in reaching the forward bearing on the pinion shaft, and in this particular design an oil passage is drilled through the wall supporting the rear bearing. The center housing should be provided with a drain hole at its lowest point and with an oil filler hole either at the top or on the

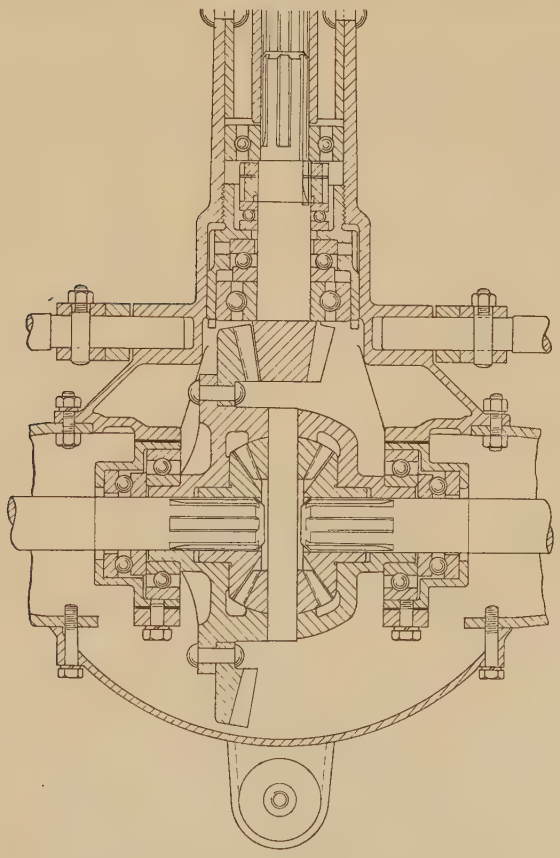


Fig. 16.—Center Housing of Ball-Bearing Rear Axle (British).

rear cover at such a height that the oil will overflow from it when the housing is filled to the proper level.

Straddle-Mounting—An axle center with a pinion having a straddle-mounting is shown in Fig. 15. With such a mounting the bearing loads naturally are considerably smaller than with the overhanging mounting; nevertheless, it is difficult

to find sufficient room for an adequate bearing at the inner end of the pinion. This design also provides means for the axial adjustment of the pinion, the adjustment being locked by a dog.

A rather different axle center from the two previously shown is illustrated in Fig. 16, this being that of the British Riley, a car of 114 in. wheelbase with 100 cu. in. engine. The housing is made of pressed steel and all bearings are ball bearings. There is a double thrust bearing with a characteristic British mounting on the pinion shaft, and two ball thrust bearings are provided on the differential, in addition to the radial bearings thereon. The radial and thrust bear-

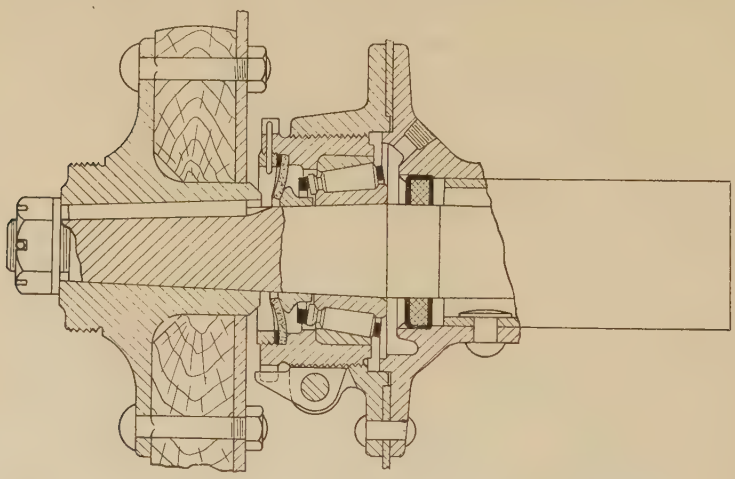


Fig. 17.—Outer End of Semi-Floating Axle with Taper-Roller Bearing.

ings on each side of the differential are carried in a mounting ring which is screwed into the differential carrier and locked by a set screw with lock washer. An oil filler is cast on the cover plate, in accordance with common British practice.

Outboard Bearing Mountings—At the outer end of a semi-floating axle there is a single bearing which is mounted in the axle housing or an extension of same. A typical design employing a taper roller bearing is shown in Fig. 17. A malleable iron bracket is secured to the end of the axle tube by riveting. A bearing hub is telescoped to this bracket and secured to it by riveting, with the brake carrier disc between.

Into the bearing hub is screwed a bearing sleeve which contains the outer race of the taper roller bearing, the inner race of which is seated on a tapered section of the axle shaft.

With a design of this type, thrust loads on the axle shaft from outside the vehicle are not taken up on the near bearing but are transmitted through the two shafts to the opposite side of the vehicle. To this end the inner abutting ends of the axle shafts are usually provided with hardened thrust buttons. Shocks in the axial direction received by the right hand shaft are taken up on the bearing at the left end of the axle. Adjustment of the roller bearings can be made by means of the threaded bearing sleeve, and the adjustment can be locked by the usual device of a toothed flange and locking dog. Outside the bearing there is an oil guard or a grease guard, consisting of a leather disc clamped between

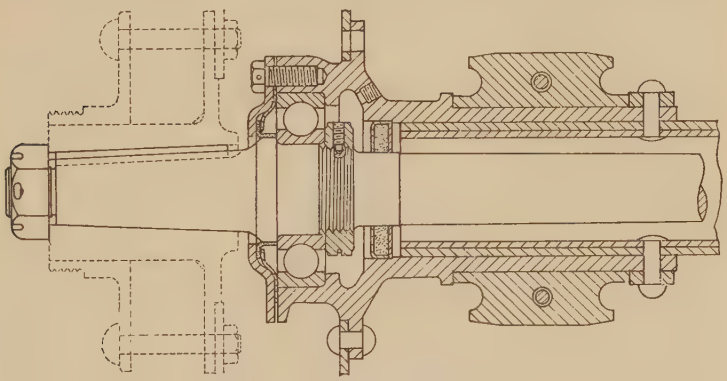


Fig. 18.—Outer End of Semi-Floating Axle with Ball Bearing.

a shoulder in the bearing sleeve and a threaded ring, and bearing against a collar on the shaft. There is an oil guard also in the bracket at the end of the axle housing, so that the bearing housing forms a lubricant chamber by itself.

The end of the axle shaft to which the wheel hub is secured is tapered and provided with a keyway, and beyond this there is a reduced threaded portion which takes a castellated nut to hold the hub in place on the shaft.

A semi-floating-rear-axle end with ball bearing is shown in Fig. 18. In this case a flanged fitting is telescoped over the end of the axle tube and secured to it by riveting. The fitting is counterbored to receive the annular ball bearing, the outer race of which is held rigidly between a flange in

the fitting and a cap bolted to it. The inner race of the bearing is held firmly between a collar on the axle shaft and a nut, and since both races are fixed, this bearing takes up all end thrust in both directions, in addition to the radial load. This design differs from the previously described one in that all of the parts at the end of the axle housing are of steel instead of malleable iron. The spring seat is held on the axle tube between a collar and a shoulder and is free to turn thereon.

Three-Quarter-Floating-Axle End—Fig. 19 shows the end of a three-quarter floating axle for a heavy passenger car. The axle tube is reduced in diameter near its end and its wall is thickened. A malleable iron combined brake bracket

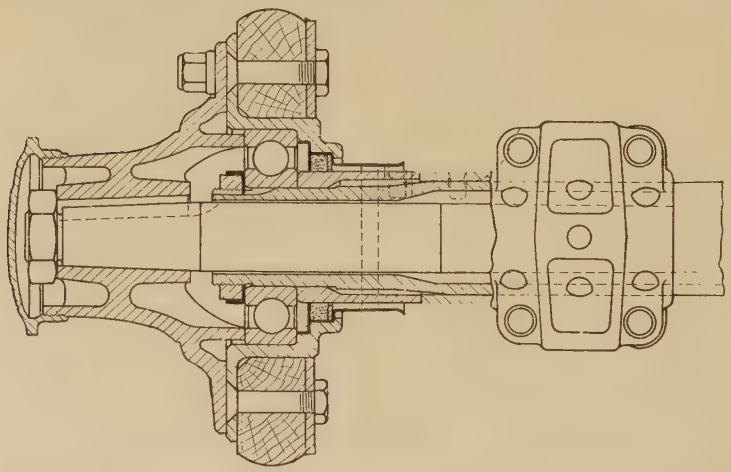


FIG. 19.—OUTER END OF PASSENGER-CAR THREE-QUARTER-FLOATING AXLE.

and spring seat is riveted to the axle tube. Beyond this there is a spacer and then comes the inner race of the ball bearing, which is clamped between the spacer and a nut. The outer race is clamped between a shoulder on the inside of the wheel hub and an internal flange cast on the driving member or clutch, which is secured to the shaft by keying and to the wheel hub by bolting.

Spiral Bevel Gears—As already pointed out, there are two types of bevel gears available for final drives, the straight and the spiral bevel gear, but the latter is used almost exclusively because of its inherent advantages for this service.

The spiral bevel gear final drive was introduced in 1913

by the Packard Motor Car Co. Spiral bevel gears with gear axes at right angles bear the same relation to straight bevel gears as helical spur gears with parallel axes to straight spur gears. Their chief advantage is their noiseless operation at all speeds, but they have a number of other important advantages which together were responsible for their almost instant popularity. These advantages are more or less inter-related. For instance, with spiral bevel gearing a smaller minimum number of teeth can be used than with straight bevel gearing. What limits the minimum number of pinion teeth in straight bevel gearing is the fact that as the number of teeth is decreased the non-uniformity of motion, and consequently the

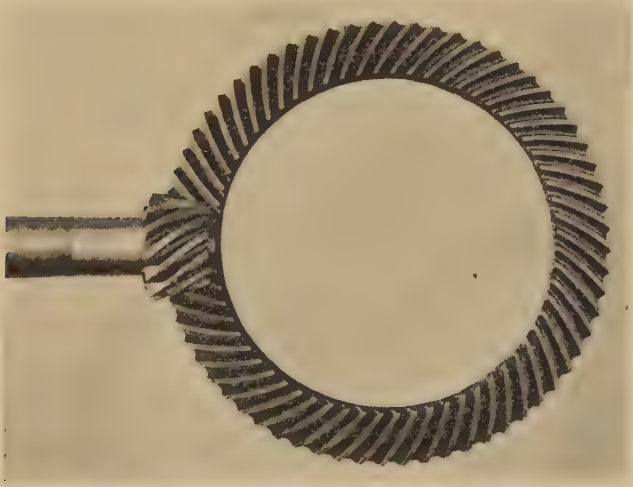


FIG. 20.—A PAIR OF SPIRAL BEVEL GEARS.

noise, increases. But spiral bevel gearing is inherently far more silent, hence this limitation is practically eliminated and pinions with a smaller number of teeth may be used.

Minimum Number of Teeth—High speed engines, especially those of moderately powered cars, require a high gear reduction and the spiral bevel gear has made it possible to obtain this higher reduction in a single step without running the risk of non-uniform and noisy tooth action.

Spiral bevel drives with pinions of eight teeth are entirely practical and these permit of obtaining any gear ratio that may be needed for passenger cars. In the case of such small numbers of teeth the pinion must be made integral with its

shaft. As regards strength, it is believed that a spiral bevel pinion of a given pitch diameter and cut with teeth of a certain diametral pitch will safely transmit the same power at a certain speed as a straight bevel pinion with the same pitch diameter and diametral pitch; this notwithstanding the fact that the normal load on the teeth is considerably greater than with the straight bevel gear. If the power transmitted is the same, the tangential force will be the same in the two cases. On the other hand the end thrust is much greater with the spiral than the straight pinion, and the normal tooth load usually figures out about 15 per cent higher in the case of the former. Probably the chief reason for the greater strength of the spiral pinion is that, since there is always pitch line contact, there can be no non-uniform motion to cause heavy extra strains. There is, however, another reason for the greater strength of a spiral bevel pinion, and that is that it has more teeth in contact at one time. If the spiral advance corresponding to the width of face is greater than the circular pitch, then the arc of action includes always at least one more tooth than in a similar straight bevel set. The two outer teeth will be in contact over only a part of their length, but as far as breakage is concerned practically their whole strength counts.

Calculation of Blanks—The calculation of the blanks for the pinions and gears partakes of the methods used for calculating the blanks for straight bevel gears and helical spur gears respectively. Thus, for instance, the pitch diameter is calculated by the same equation as used in the case of helical spur gears, viz.,

$$d_p = \frac{N}{p_n \times \cos \alpha},$$

where N is the number of teeth; p_n, the normal diametral pitch and α the angle which the tooth element makes with the pitch cone element (angle of spiral). Spiral bevel pinions are usually made with a long addendum. In standard spur gears the addendum is made equal to $0.3183\ p_c = 1/p_d$, and the dedendum, $0.3683\ p_c = 1.157/p_d$, the latter being the sum of a working depth of $0.3183/p_c$ below the pitch circle and a clearance of $0.05\ p_c = 0.157/p_d$. Therefore, the total working depth of $0.6866\ p_c\ (= 2/p_d)$ extends equally above and below the pitch circle. Now it is known that in a pinion with a small number of teeth there is a tendency to undercutting and consequent weakening of the pinion teeth. To obviate this it is customary in spiral bevel pinions to have most of the working depth above

the pirch circle. The usual practice is to make the working depth above the pitch circle of the pinion 0.7 of the total working depth, and the working depth below the pitch circle, 0.3 of the total. In the gear the proportion is reversed, 0.7 being below the pitch circle and 0.3 above it.

Let it be required to lay off the blanks for a spiral bevel gear and pinion of 48 and 12 teeth respectively, 5 pitch, 30° angle of spiral with 0.7 of the working depth above and 0.3

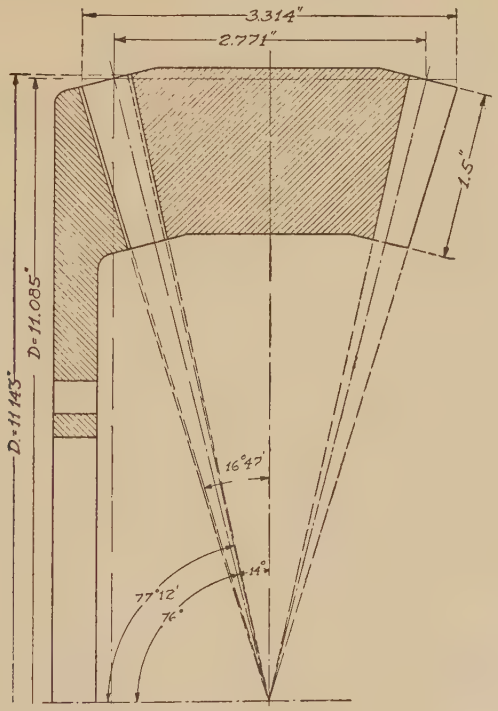


Fig. 21.—Lay-out of a Pair of Spiral Bevel Gears.

below the pitch circle in the pinion. We have in the first place for the maximum pitch diameters

$$\frac{12}{5 \times 0.866} = 2.771 \text{ ins.}$$

and

$$\frac{48}{5 \times 0.866} = 11.085 \text{ ins.}$$

The blanks are laid off to scale in Fig. 21.
The total working depth of the 5 pitch teeth is

$$\frac{2}{5} = 0.4 \text{ in.}$$

hence, in the pinion the working depth above the pitch circle or the addendum is

$$0.7 \times 0.4 = 0.28 \text{ in.}$$

and the working depth below the pitch circle

$$0.3 \times 0.4 = 0.12 \text{ in.}$$

The clearance is

$$\frac{0.157}{5} = 0.0314 \text{ in.}$$

The pitch angle of the pinion is such that

$$\text{tangent pitch angle} = \frac{2.771}{11.085} = 0.25$$

$$\text{pitch angle} = 14°.$$

The face angle of the pinion is greater than the pitch angle by an angle such that its tangent is

$$\frac{0.28}{\sqrt{1.385^2 + 5.543^2}} = 0.0487,$$

and the angle is 2° 47′.
Hence, the face angle is 16° 47′.
The face diameter of the pinion is

$$2.771 + (2 \times 0.28 \times \cos 14°) = 3.314.$$

The pitch angle of the gear is the complement of the pitch angle of the pinion or

$$90° - 14° = 76°$$

and the angle which the addendum adds to the pitch angle is such that its tangent is

$$\frac{0.12}{\sqrt{1.385^2 + 5.543^2}} = 0.0208$$

and the angle is 1° 12′,
hence, the face angle is

$$76° + 1° 12′ = 77° 12′.$$

Capacity of Spiral Bevel Gears—In the Chapter on The Differential Gear the following equation for the maximum safe load on the pitch line of straight bevel gears was developed:

$$F = \frac{SLpy}{3}(1 - a^3).$$

A similar equation may be used for spiral bevel gears. Unfortunately the profile factors y seem never to have been determined for pinions with long addendum teeth. However, for the small numbers of teeth (around 10) used in the pinions of rear axle spiral bevel drive gears these factors vary substantially in direct proportion to the number of teeth, and we may therefore write

$$F = cLpn\,(1 - a^3),$$

where c is a constant. Application of this equation to a number of successful designs shows that c has an average value of 250. Therefore, for rear axle drive gears of the spiral bevel type, made of No. 2320, No. 3120 or a similar steel, case-hardened, with long addendum pinion teeth and a spiral angle of about 30°, the safe load on the pitch circle, based on the maximum engine torque multiplied by the transmission low gear reduction ratio, is

$$F = 250Lpn\,(1 - a^3) \text{ lbs.},$$

where L = the cone distance in inches;
p = the normal circular pitch;
n = the number of pinion teeth;
a = the proportion of the distance measured along a pitch element from the inner end of a tooth to the apex, to the distance from the outer end of a tooth to the apex.

Direction of Thrust Loads—When a car is being driven forward, the propeller shaft and bevel pinion turn right-handedly, while when the car is being backed the bevel pinion turns left-handedly. Therefore, a right-hand pinion tends to draw into the gear when the car is being driven forward, as a result of the curvature of the teeth. The other causes of end thrust, viz., the taper of the pinion cone and the pressure angle, tend to force the pinion out of the gear. Of these forces the first mentioned, that is, that due to the spiral angle, always predominates, and the net end thrust on the

shaft of a right-hand pinion is in the direction toward the gear center and equal to the difference between the end thrust due to the curvature of the teeth on the one hand and that due to the pinion cone angle and the pressure angle on the other.

In backing, the end thrust due to the curvature of the teeth of a right-hand spiral pinion is away from the gear center and in the same direction as the end thrust due to the cone angle and pressure angle. The resultant is, therefore, in the direction away from the gear center and equal to the sum of the end thrusts due to tooth curvature, cone angle and pressure angle, respectively. Evidently, therefore, with a right-hand pinion the end thrust is a maximum when the car is being backed.

With a left-hand pinion all end thrusts add together for forward drive and are in the direction away from the center of the gear, while for the reverse drive the end thrust, though still in the same direction, is equal in amount to the difference between that due to the spiral angle on the one hand and to the cone angle and pressure angle on the other. The maximum end thrusts are the same whether a right- or left-hand pinion is used. As the car is being driven forward most of the time, the right-hand pinion seems to have the advantage, but some designers prefer the left-hand pinion, because heavy thrust loads in the direction away from the gear center can be accommodated more readily than those in the opposite direction.

Bearing Loads with Spiral Bevel Gears—We found in the chapter on The Differential Gear that the center of load distribution in a bevel gear is located at a distance mL from the apex of the pitch cone, where L is the cone distance and m is given by the equation

$$m = \sqrt{1 + \frac{n^2}{2}} - n,$$

n being the proportion of the gear face to the cone distance. A tabulation of values of m for different values of n within the usual range was given on page 158 in that chapter.

In determining the bearing loads we have to first determine the tangential force or tooth load on the pinion and its location. The mean effective pitch diameter d_p of the pinion is found by multiplying the maximum or nominal pitch diameter by the value of m. We then get for the tangential force

$$F = \frac{2A}{d_p}T.$$

The normal load on the tooth contact surface is, of course, considerably greater than the tangential force, owing to the inclination of the tooth elements to the pitch cone elements (angle of spiral) on the one hand, and to the inclination of the tooth flank (pressure angle) on the other. If we designate the angle of spiral by α and the pressure angle of the tooth by β, the normal pressure on the tooth is

$$P = \frac{F}{\cos \alpha \cos \beta},$$

as may be readily seen from Fig. 22 in which AB represents the tangential force on the pinion, AC the force in a plane tangent to the pitch cone and perpendicular to the tooth element, and CD the force normal to the contact surfaces.

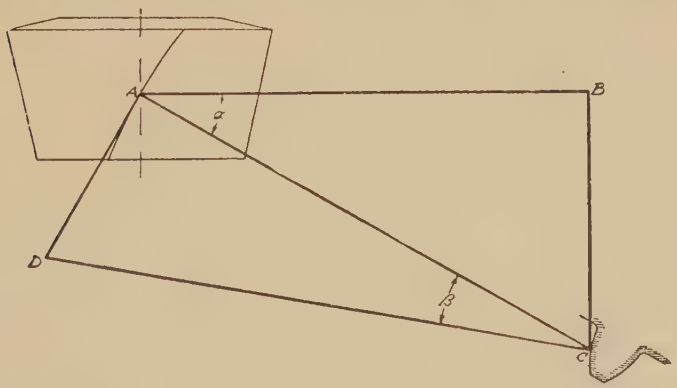


FIG. 22.—SHOWING RELATION BETWEEN TANGENTIAL FORCE AND NORMAL TOOTH PRESSURE.

The problem now is to resolve the pressure P into a component parallel to the pinion axis (thrust load) and another component perpendicular to the pinion axis (radial load).

We first find the components of the tooth pressure along an element of the pitch cone and perpendicular to that element, respectively. In Fig. 23 is shown a plan view of a right-hand pinion supposed to rotate right-handedly. The horizontal arrow represents the tangential force F on the inclined tooth and the vertical arrow the resulting pressure along the pitch cone element. It will be seen that

$$\frac{C_e}{F} = \tan \alpha, \quad \text{hence } C_e = F \tan \alpha.$$

This pressure along the pitch cone element can again be resolved into two components, as shown in Fig. 24, one parallel to the pinion axis and the other perpendicular thereto. It is here necessary to take account of the direction of the forces and we will call axial forces in the direction from the small to the large end of the pinion, or away from the apex of the cone, positive, and those in the opposite direction, negative. Radial forces from the point of contact toward the axis will be called positive and those oppositely directed, negative.

Resolving the force along the pitch cone element we get for the axial component

$$C_a = -F \tan \alpha \cos \delta,$$

and for the radial component

$$C_r = F \tan \alpha \sin \delta.$$

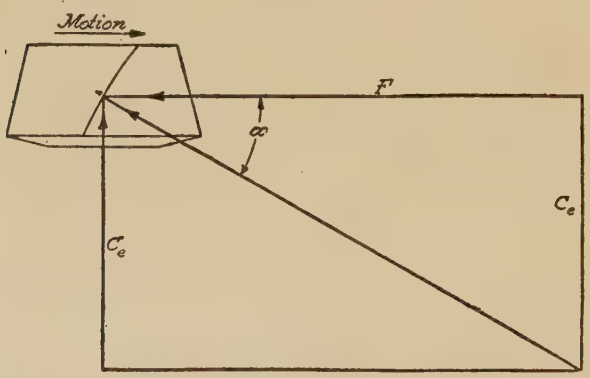


FIG. 23.—SHOWING RELATION BETWEEN TANGENTIAL FORCE AND PRESSURE ALONG PITCH-CONE ELEMENT.

We next take up the component of the normal tooth pressure perpendicular to the pitch cone element through the point of contact. From Fig. 25 it can be seen that this component C_p is equal to

$$P \sin \beta = \frac{F \sin \beta}{\cos \alpha \cos \beta} = F \frac{\tan \beta}{\cos \alpha}.$$

This component perpendicular to the pitch cone element also may be further resolved into axial and radial components, as illustrated in Fig. 26, the axial component being

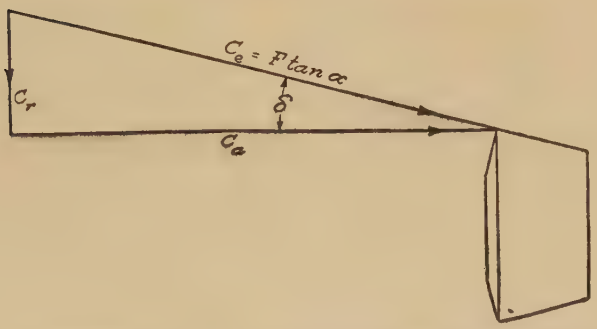


FIG. 24.—PRESSURE ALONG PITCH-CONE ELEMENT RESOLVED INTO AXIAL AND RADIAL COMPONENTS.

$$C_{aa} = F \frac{\tan \beta \sin \delta}{\cos \alpha},$$

and the radial component

$$C_{rr} = F \frac{\tan \beta \cos \delta}{\cos \alpha}.$$

We now add like components of the forces along the pitch cone element and perpendicular to that element respectively and obtain for the end thrust on a right-hand spiral bevel pinion turning right-handedly (forward drive)

$$L_a = F \left(- \tan \alpha \cos \delta + \frac{\tan \beta \sin \delta}{\cos \alpha} \right)$$

and for the radial load on such a pinion

$$L_r = F \left(\tan \alpha \sin \delta + \frac{\tan \beta \cos \delta}{\cos \alpha} \right)$$

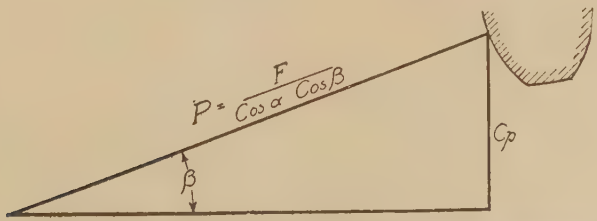


FIG. 25.—COMPONENT OF NORMAL TOOTH PRESSURE PERPENDICULAR TO PITCH-CONE ELEMENT.

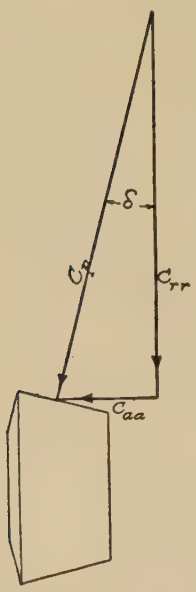


FIG. 26. — PRESSURE PERPENDICULAR TO PITCH-CONE ELEMENT RESOLVED INTO AXIAL AND RADIAL COMPONENTS.

These same equations apply to the case of a left-hand pinion turning left-handedly (reverse motion). If a left-hand pinion turns right-handedly (forward motion), the component along the pitch cone element is in the direction away from the apex of the cone and the sign of its axial component becomes positive. The same applies to a right-hand pinion turning left-handedly. However, the radial component of the force acting along the pitch cone element is in this case directed from the axis through the point of contact and is, therefore, negative. We, therefore, have for the axial and radial forces on a left-hand spiral bevel pinion turning right-handedly (forward drive) or a right-hand spiral bevel pinion turning left-handedly (reverse drive).

$$L_a = F\left(\tan\alpha\cos\delta + \frac{\tan\beta\sin\delta}{\cos\alpha}\right)$$

and

$$L_r = F\left(-\tan\alpha\sin\delta + \frac{\tan\beta\cos\delta}{\cos\alpha}\right).$$

Axial and Radial Loads on Gear—Since action and reaction are equal and opposite, the axial load or end thrust on the gear is equal and opposite in direction to the radial load on the pinion, and the radial load on the gear is equal and opposite in direction to the axial load on the pinion. There is, therefore, no need of separately calculating the gear-bearing loads. We may summarize our equations as follows:

Right-Hand Pinion Turning Right-Handedly.
Left-Hand Pinion Turning Left-Handedly.

End Thrust on Pinion, Radial Load on Gear.

$$L_1 = F\left(-\tan\alpha\cos\delta + \frac{\tan\beta\sin\delta}{\cos\alpha}\right).$$

Radial Load on Pinion, End Thrust on Gear.

$$L_2 = F\left(\tan\alpha\sin\delta + \frac{\tan\beta\cos\delta}{\cos\alpha}\right).$$

Right-Hand Pinion Turning Left-Handedly.
Left-Hand Pinion Turning Right-Handedly.

End Thrust on Pinion, Radial Load on Gear.

$$L_3 = F\left(\tan\alpha\cos\delta + \frac{\tan\beta\sin\delta}{\cos\alpha}\right).$$

Radial Load on Pinion, End Thrust on Gear.

$$L_4 = F\left(-\tan\alpha\sin\delta + \frac{\tan\beta\cos\delta}{\cos\alpha}\right).$$

in which

F is the tangential force at the mean effective pitch radius of the pinion; α, the angle of spiral; β, the pressure angle of the teeth; δ, the pinion pitch angle.

The Hypoid Gear—In 1926 a new form of gearing, known as the hypoid gear, was introduced for final drives on automobiles. A hypoid gear is similar to a bevel gear, except that the axis of the pinion is offset from the axis of the gear. In automobile drives the axis of the pinion is placed below the axis of the gear, and the immediate object in view when replacing the spiral bevel by the hypoid gear was to take advantage of this lowering of the pinion shaft and the consequent lowering of the propeller shaft, to place the frame and body floor of the car lower, thus increasing its stability and adding to its safety at high speed. A pair of hypoid gears is illustrated in Fig. 27.

Machinery for cutting hypoid gears was developed by the Gleason Works of Rochester, N. Y., and the first concern to use such gears on automobiles was the Packard Motor Car Co. of Detroit. The offsets of hypoid gears have been standardized at 1½, 2, 2½, 3 and 3½ in. The direction of the offset determines the hand of the spiral. When the pinion is below the gear axis, as in automobile drives, the spiral of the pinion must be made left-hand. With this arrangement the end thrust due to forward drive tends to force the pinion away from the gear, rather than to wedge it into same, which is an advantage, since thrust in the direction away from the apex of the pinion pitch cone can be more readily taken care of. It is also claimed that with this arrangement the tooth contact on the driving side is more intimate, especially in pairs with a moderate ratio, as used on high-powered passenger cars.

The offset of the axes in a hypoid set is the cause of an endwise sliding motion, and the teeth must be given a form in the lengthwise direction which will permit of such motion.

This condition imposes restrictions on the pitch lines which are not present with ordinary bevel gears. The same as in worm gears, the circular pitches of the gear and pinion of a hypoid set are not equal, but are proportional to the peripheral speeds of these members. The normal pitch, of course, is the same in both. The pitch diameter is proportional to the circular pitch and the number of teeth. The spiral angle of the pinions is made 35° or 40°, while the spiral angle of the gear is held between 5° and 15°. Owing to the large spiral angle, the pitch diameter of the hypoid pinion is from 15 to 20 per cent greater than that of a spiral bevel pinion of the same normal pitch and number of teeth, and this makes the hypoid pinion much stronger Another advantage of this feature is that large ratios, which it is difficult to obtain with spiral bevel gears because the pinions become too small (and therefore too weak), are practical with hypoid gears.

Owing to the fact that the spiral angle of the gear is only small (between 5° and 15°) the tooth load on the gear is not much greater than the tangential force, in which respect the hypoid also compares favorably with the spiral bevel. Opposite sides of the teeth of hypoid gears have different shapes, the pressure angle being made 17½° on the driving side and 20° on the reverse side, that is, on the concave and convex sides of the pinion teeth.

With the same reduction ratio and the same amount of power to be transmitted the same numbers of teeth as with spiral bevel gears would be chosen, and the pitch diameter of the gear—and hence the road clearance—would be the same, while the pitch diameter of the pinion would be greater and the pinion would be stronger. One disadvantage of the hypoid gear is that the bearing in back of the pinion cannot be brought as close to it as in the case of spiral bevel gears, and the overhang of the pinion therefore will be greater. This calls for a particularly rigid mounting of the pinion. The overhang of the center of the pinion should not exceed one-third the center distance between the pinion supporting bearings, and great rigidity should also be aimed at if the pinion has a straddle mounting. It is also quite desirable to have the gear very rigid, so it cannot spring away from the pinion appreciably under the influence of the thrust on it, and it is recommended to make the ring gear without a web and to stiffen the flange on the differential housing to which it is fastened by suitable radial ribs.

The pitch diameter of a hypoid gear is equal to the product of its number of teeth into its circular pitch divided by 3.1416. This same rule also applies to the pinion, whose

circular pitch is different from that of the gear, however. In each case the circular pitch is equal to the normal circular pitch—which is the same for both pinion and gear—divided by the cosine of the angle of spiral.

Bearing Loads with Hypoid Gears—With hypoid gears the bearing loads on the bearings of the pinion shaft and the gear must be calculated separately, as the axes of pinion and gear do not intersect and the thrust load on one therefore is not equal to the radial load on the other, as in the case of bevel gears. The pressure angle β, of course, is the same for both the pinion and the gear, but the spiral angle $\propto$ and the pitch angle δ are different for the pinion and gear, and we will use sub-letters p and g with $\propto$ and δ to designate that

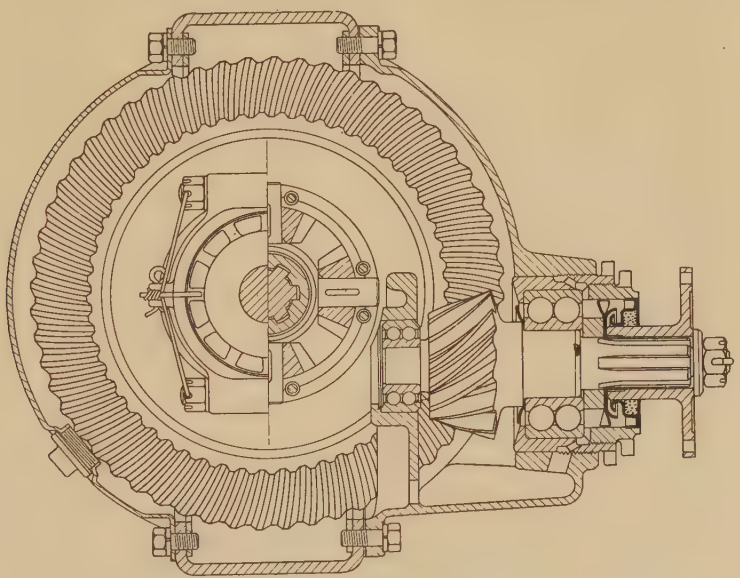


FIG. 27.—HYPOID GEARS MOUNTED IN REAR AXLE.

they apply to the pinion and gear respectively. In the following equations for axial and radial loads the plus sign denotes that the load is away from the cone center, while the minus sign denotes that the load is in the direction toward the center.

Left-hand pinion and right-hand gear, pinion turning right-handedly—or right-hand pinion and left-hand gear, pinion turning left-handedly.

Pinion Loads

$$L_a = \frac{F}{\cos \alpha_g} (\sin \alpha_p \cos \delta_p + \tan \beta \sin \delta_p)$$

$$L_r = \frac{F}{\cos \alpha_g} (- \sin \alpha_p \sin \delta_p + \tan \beta \cos \delta_p)$$

Gear Loads

$$L_a = \frac{F}{\cos \alpha_g} (- \sin \alpha_g \cos \delta_g + \tan \beta \sin \delta_g)$$

$$L_r = \frac{F}{\cos \alpha_g} (\sin \alpha_g \sin \delta_g + \tan \beta \cos \delta_g).$$

Left-hand pinion and right-hand gear, the pinion turning left-handedly, or right-hand pinion and left-hand gear, the pinion turning right-handedly.

Pinion Loads

$$L_a = \frac{F}{\cos \alpha_g} (- \sin \alpha_p \cos \delta_p + \tan \beta \sin \delta_p)$$

$$L_r = \frac{F}{\cos \alpha_g} (\sin \alpha_p \sin \delta_p + \tan \beta \cos \delta_p).$$

Gear Loads

$$L_a = \frac{F}{\cos \alpha_g} (\sin \alpha_g \cos \delta_g + \tan \beta \sin \delta_g)$$

$$L_r = \frac{F}{\cos \alpha_g} (- \sin \alpha_g \sin \delta_g + \tan \beta \cos \delta_g).$$

Determination of Bearing Sizes—After the components of the tooth pressure parallel to the gear axis and in a plane perpendicular to that axis have been determined by means of the equations given in the foregoing, the next step is to divide the radial load between the two bearings on which it is taken. This can be done by the methods explained in the chapter on Transmissions. If the pinion is straddle-mounted the sum of the loads on the two bearings is equal to the component of the tooth load in the plane at right angles to the

pinion axis, the load on each bearing being inversely proportional to its distance from the center plane of the tooth load. If, on the other hand, the pinion overhangs its bearings, the load on the nearby bearing is greater than the tooth load, and the load on the farther bearing is equal to the difference between the load on the nearby bearing and the tooth load. In this case, also, the loads on the two bearings are inversely proportional to their distances from the center plane of the tooth load.

The load on any one bearing varies greatly according to whether the engine power is transmitted through the first, second, third or fourth gear of the transmission. If the rating applied to the bearings is that usually employed in American ball bearing practice, which assures a life equal to that of the average car if the rating of transmission bearings is equal to the calculated load on them in second gear, then the ratings of the bearings on the bevel gear pinion and on the differential should be about twice the calculated load on these bearings for full engine power and direct drive. If end thrust is taken on annular ball bearings it is advisable to make a separate calculation for end thrust capacity required. Most manufacturers of ball bearings give the thrust load capacities of their different types of bearings in terms of the radial load capacities of same.

Wheel Bearings—The loads on the bearings in the wheels depend directly on the loads supported by the wheels. The actual bearing load comprises a vertical component equal to the static load on the wheel, and a horizontal component equal to the driving force, which attains a maximum value equal to about 0.6 that of the load on the wheel. The actual bearing load then is 1.165 times the wheel load. This proportion is reached only when the driving effort is sufficient to slip the wheels. In ordinary driving in high gear the bearing load is only about 2 per cent greater than the wheel load. The wheel bearings should be so chosen that their rated capacity, if determined as above explained, is about 200 per cent of the calculated load.

Axle Truss—In the early years of the automobile industry a good deal of use was made, in connection with built-up axles, of trusses for rear axle housings, as illustrated in Fig. 28, but this practice has been almost entirely discontinued. The middle of the truss was generally retained between projections on the driving gear housing and the ends were secured to lugs which were formed on the fittings at the ends of the axle tubes (the spring seats or the brake spider). The truss could be tightened either by nuts on it

at both ends outside the brake spider, or by means of turnbuckle, as shown in Fig. 28, in which latter case the rods were pinned to lugs on the axle fittings. The threaded ends were frequently upset, so they were of the same strength as the solid portion of the rods. In view of the fact that trusses are rarely used at present it is not necessary to go into the details of their design, but it may be pointed out that they are capable of greatly relieving that portion of the axle housing between their anchorages of bending stresses, and therefore permit of lighter axle housings. There are probably a number of reasons for their non-use. An axle housing which has sufficient rigidity in itself to stand up under the loads upon it is naturally more secure than one depending upon an extraneous member which may be readily damaged in service or be incorrectly adjusted by an unskilled operator. With pressed steel housings the central portion is not very stiff and might

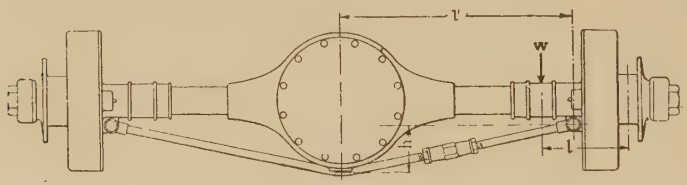


FIG. 28.—TRUSSED REAR AXLE.

be deformed when acting as a prop for the truss. Finally, with cast steel or malleable iron housings the minimum sections which can be successfully cast are so heavy that the housing has easily sufficient rigidity for the loads upon it. The built-up type of rear axle housing, which is weakest where the tubes enter the hubs on the cast center housing, benefits most from a truss.

Rear Axle Torsion and Thrust—All of the forces and moments or couples acting on the rear axle are productive of reactions. There are two moments or torques, namely, the torque on the driving pinion shaft and the torque on the axle shafts, and each torque gives rise to a torque reaction which is equal to it in magnitude and opposite to it in direction. Referring back to Fig. 11, the driving bevel pinion, when looked at from the front of the car, turns right-handedly and presses down upon the teeth of the bevel gear. To enable it to exert this downward pressure on the gear teeth, it presses upward on the bearing supporting its shaft. The direct pressure of the pinion teeth on the gear teeth and of the pinion

shaft against its bearing or bearings constitute a couple which is balanced by another couple due to an upward reaction on the left rear wheel and a downward reaction on the right rear wheel. The effect of the torque on the pinion shaft is to increase the load on the left rear wheel and to decrease the load on the right rear wheel. The amount of weight removed from one and added to the other wheel can be readily found by merely dividing the pinion shaft torque in lb.-in. by the wheel tread in in. This transfer of weight reaches its maximum when the rear-axle torque is so great that the wheels approach the slipping point on dry pavement, in which case, in a car with standard tread, it amounts to about 7 per cent of the weight originally on each wheel. The balance of the car is therefore not disturbed and it is not necessary to pay any special attention to this (propeller shaft) torque reaction.

It is different with the reaction to the torque on the rear axle, because this is greater in the proportion of the rear axle reduction ratio. When the rear wheels are being turned around or urged around in one direction with a given torque, the axle housing tends to turn in the opposite direction with the same torque. Since the axle housing is freely supported within the wheel hubs or on the axle shafts, on anti-friction bearings, there is nothing to prevent it from turning except the members which connect it to the vehicle frame. One or more of these connections must be so designed as to be capable of resisting strong torques. The magnitude of this turning moment on the rear axle housing has been often misjudged. A good idea of it can be obtained by reflecting that in order to balance it at the end of an arm equal in length to a wheel radius, a force is required which is equal to the propelling effort exerted by the two wheels.

In addition to the torque, the driving force must be provided for. All of the driving effort originates at the contact of the driving wheels with the ground, whereas much of the resistance to motion originates at the front wheels and on the surfaces of the body (air resistance), and the rear axle virtually must push the car body and front axle forward. In the opposite case, when brakes are applied to the rear wheels, the driving effort originates mainly in the chassis (due to its inertia) while the rear axle tends to remain behind, and then the chassis pulls the rear axle instead of being pushed by it. Hence it is necessary to have a connection or connections between the axle and the chassis frame which can transmit forces in both directions, or in other words, transmit both a pull and a push.

The Hotchkiss Drive—In many passenger cars both the rear axle torque and the propelling force and brake pull are taken up by the rear chassis springs, the method of construction making this possible being known as the Hotchkiss drive. Cars with Hotchkiss drive always have half-elliptic springs, which are rigidly secured to spring seats on the axle housing and are pinned to the chassis frame at their forward end. The Hotchkiss drive calls for springs that are substantially flat when under load. The springs naturally yield under severe torsion, which is of advantage in that it tends to protect the transmission members, the wheels and tires against shock. Of course, the springs are rather severely stressed by their double duty of cushioning road shocks and counteracting rear axle torque, particularly the main leaves, and these should always be made of specially good spring steel.

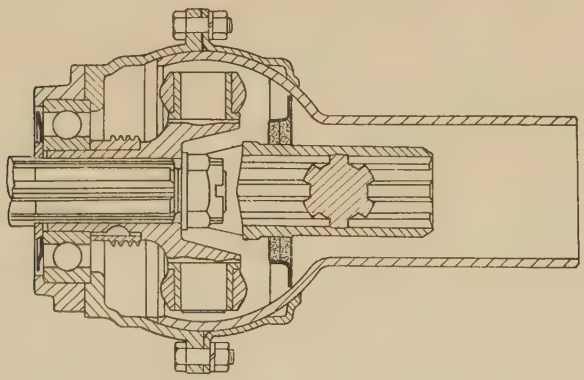


FIG. 29.—BALL-AND-SOCKET SUPPORT FOR FORWARD END OF TORQUE TUBE.

Torque Tubes—Another means of taking care of the torque reaction and the driving thrust is the torque tube, which surrounds the propeller shaft. The torque tube is fitted rigidly to the rear axle center housing and does away with the use of a universal joint at the rear end of the propeller shaft, which is connected to the bevel pinion shaft by a splined and pinned coupling. At the forward end the torque tube is supported from either a cross-member of the frame or from the rear end of the transmission case. Two methods of support are in vogue, the ball-and-socket method and the yoke method.

A method of supporting the forward end of the torque

tube by a ball-and-socket joint is illustrated in Fig. 29. One-half of the socket is a casting which bolts to the rear end of the transmission housing, and also serves to hold the outer race of the main shaft rear bearing in place. The other half is a sheet metal drawing, both halves being flanged and held together by bolts. The ball is made of sheet metal and is formed integral with a sleeve fitting over the forward end of the torque tube. The universal joint is located concentric with the ball-and-socket and the chief advantage of this form of construction is that it furnishes an oil and dust-proof

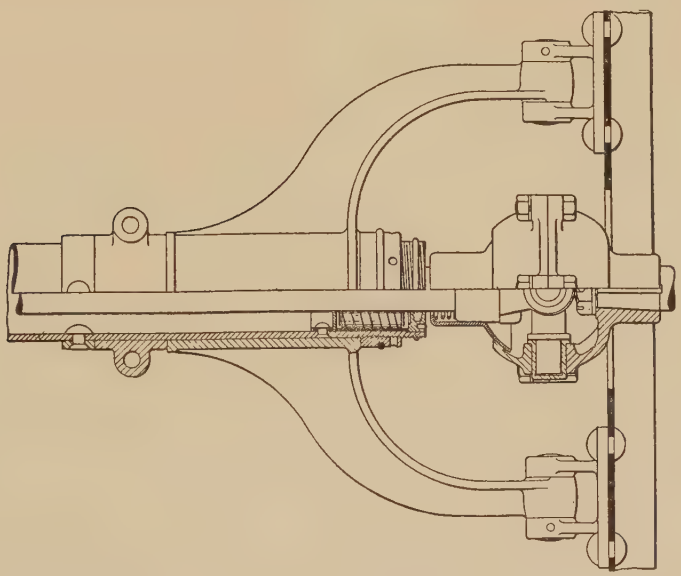


FIG. 30.—YOKED SUPPORT FOR FORWARD END OF TORQUE TUBE.

housing for the universal joint, which latter, therefore, need not be of the enclosed type. In this particular design the space within the socket communicates with the interior of the transmission housing, and the same supply of lubricant serves for both compartments. It will be noticed that there is an oil guard in the ball which tends to prevent oil from entering the torque tube.

A method of supporting the forward end of the torque tube by means of a yoke is illustrated in Fig. 30. In order that the rear axle may be able to rock freely in a vertical plane, to accommodate itself to road unevennesses, the yoke

must swivel on the forward end of the torque tube. The joints of the yoke and the abutting surfaces of its hub should be liberal in size, and means of lubrication must be provided. The yoke can be drop forged, and its arms are usually of T-section. The axis of the hinged joint must, of course, coincide with the center of the universal joint, so that spring action will not add to the loads on the bearing of the propeller shaft in the forward end of the torque tube. This bearing, as illustrated, is generally of the flexible roller type. A similar bearing is used when the forward end of the torque tube is supported by a ball-and-socket type of joint.

The loads on the ball-and-socket or yoke bearings for any condition of operation can be readily determined by merely dividing the rear axle torque in lb.-in. for that condition of operation by the distance of the center of the bearing in question from the rear axle axis, in inches.

Diagonal Brace Rods—In any vehicle in which the driving thrust and brake pull are taken on a central torque tube and the rear springs are connected to the car frame by shackles throughout, the rear axle is subjected to important bending moments when one rear wheel strikes a large road obstruction or only one of the rear brakes is effective. Such moments are generally taken care of by providing diagonal braces between the spring seats or brake supports on the rear axle housing and the forward end or an intermediate point along the length of the torque tube. A typical rear axle construction with diagonal brace rods is shown in Fig. 31. The braces are generally tubular and may be provided with pinned joints at both ends or they may have a pinned joint at the rear end while at the forward end a threaded rod welded to the tube may extend through a hole in a lug on a torque tube fitting and provided with nuts on both sides of the lug.

The spring seats and brake supports also are integral parts of the rear axle but they will be discussed in the chapters on springs and brakes, respectively.

Torque Arms—Instead of taking up the torque reaction on a tube surrounding the propeller shaft, it may be taken up on an arm which is either rigidly secured to the rear axle center housing or else has a pivot joint thereon, generally the latter. Fig. 32 shows such a torque arm. Since the bending moment on the torque arm decreases in proportion to the distance from the rear axle axis, the arm proper is preferably made of pressed steel of channel section, the depth of the channel decreasing with the distance from the rear axle. Forged fittings can be secured to the pressing by riveting or bolting.

At its forward end the torque arm is supported from a frame cross-member. The joint at this point is generally of such a nature that the driving thrust and brake drag are not taken by the torque arm. This means that the torque arm is supported from the frame through some sort of link. Formerly this connection was generally metallic, and the end of the torque arm was supported between coiled springs, so that it was cushioned in both directions. On passenger cars

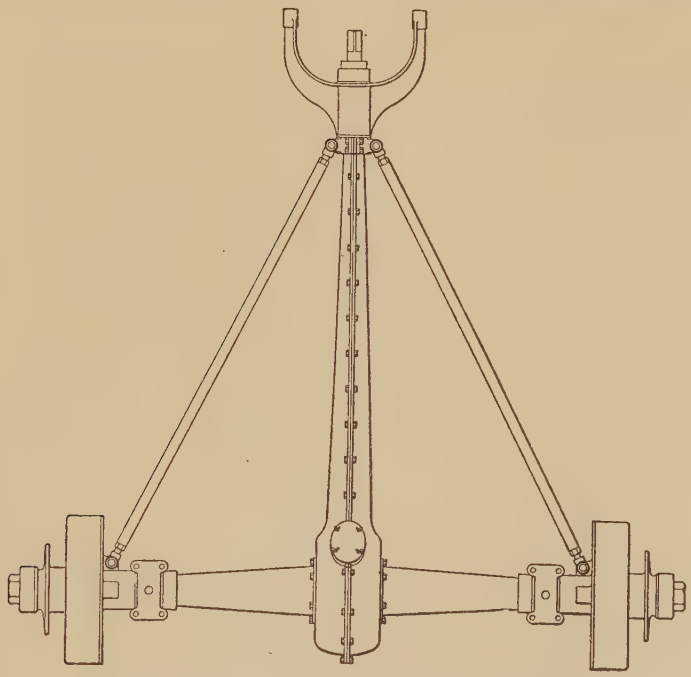


FIG. 31.—AXLE WITH DIAGONAL BRACE RODS.

the metallic connection proved a source of trouble, however, as it is difficult to get at for lubrication, and wear resulted in an unpleasant noise. The more recent practice in connection with torque arms therefore is to use a flexible rubberized fabric connection, which, of course, must be so designed that it will support the torque arm in both directions without allowing it any appreciable amount of play. Such a torque arm support, known as the Belflex, is illustrated in Fig. 33.

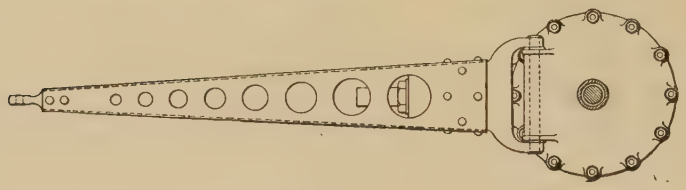


FIG. 32.—ONE FORM OF TORQUE ARM.

Radius Rods—While on the subject of members connecting the frame to the rear axle it may be well to also refer to radius rods, though these are hardly ever used for passenger cars at the present. Radius rods are designed to take the driving thrust and brake pull only, and as a rule one such rod extends from the frame to the rear axle housing under each frame side member. Radius rods are jointed at both ends, so they cannot take the rear axle torque, and preferably the joints should be of a universal type, so that they and the rods will not be unduly strained by lateral motion of the axle relative to the frame as may result when striking an obstruction when skidding. Fig. 34 shows a typical radius rod design.

Rear axle torque and driving thrust therefore may be taken care of in the following manners: Both may be taken on the chassis springs and both may also be taken on a torque

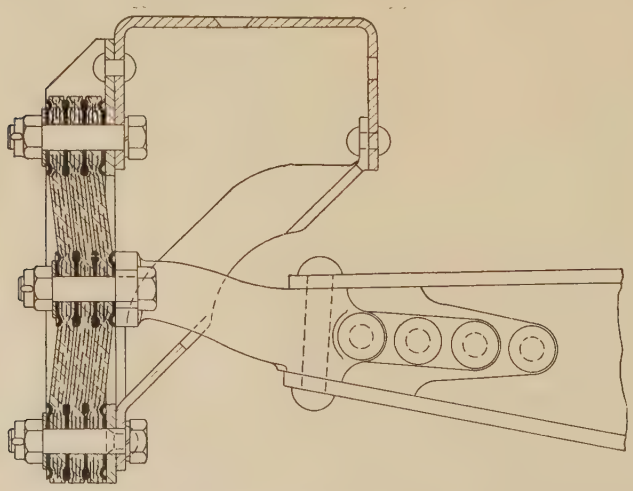


FIG. 33.—NON-METALLIC TORQUE-ARM SUPPORT (Belflex).

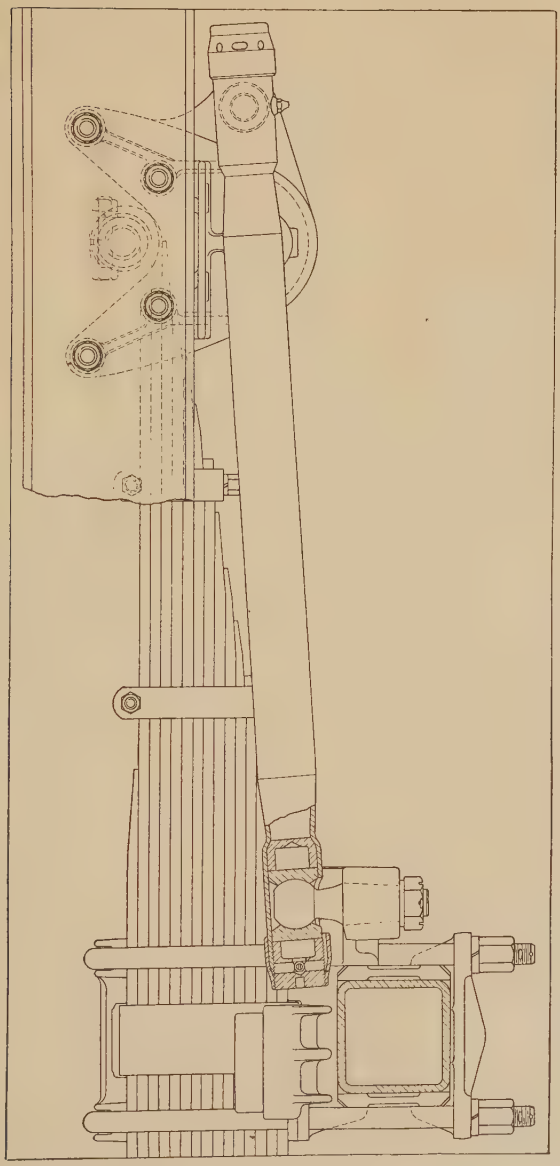


FIG. 34.—TUBULAR RADIUS RODS EXTENDING BETWEEN BALL STUDS MOUNTED ON SPRING BRACKETS.

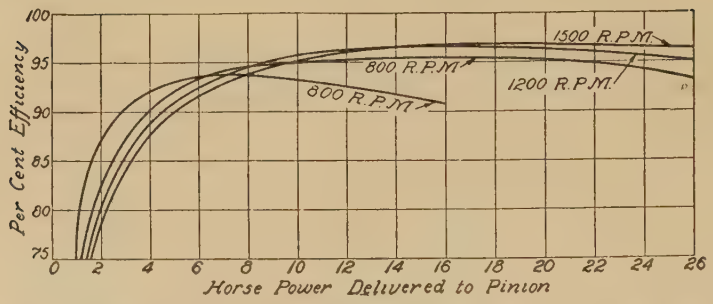


FIG. 35.—EFFICIENCY OF BEVEL-GEAR-DRIVEN, FULL-FLOATING REAR AXLE.

tube. In addition the torque may be taken on a torque tube or a torque arm and the driving thrust either on the springs or on radius rods.

Bevel Gear Efficiency—Some tests on the efficiency of bevel gear driven rear axles were made at the plant of the H. H. Franklin Mfg. Co. and the results are given in Fig. 35. Two absorption dynanometers were used, one connected to each of the two axle shafts, and the differential was locked. While these test figures refer to a rear axle with straight bevel gears, it is believed that the results would not have been materially different if the gears had been of the spiral bevel type. The axle was a full-floating design and carried bevel gears with 5 diametral pitch, 14½° pressure angle teeth, the ratio being 52/15. The gears were machined up from 3½ per cent nickel steel blanks, case hardened, and the teeth were burnished by running the gears together in a mixture of emery and oil. The length of face was 1½ in. Previous to the tests the axle had been used on a car for about 3000 miles.

It will be seen from the diagram that the maximum efficiency is about 97 per cent and that the efficiency is above 95 per cent for a considerable range in horse power transmitted and speed of revolution. During the test the axle gears were run in a bath of graphite and oil. The losses indicated by the diagram include both gear and bearing losses, but bearing losses were very small, as ball bearings were used throughout. A semi-floating axle was also tested and showed substantially the same efficiencies, except that at small loads its efficiency was slightly higher than that of the full-floating design.

CHAPTER VIII

The Worm Drive

The worm drive is used to a small extent for passenger cars and more largely for trucks and buses. For farm tractors it has been used by Ford and others. Its chief advantages are that it assures absolute silence of operation under all conditions, and that it permits of obtaining a larger reduction in a single step than is possible under automobile conditions with any other known form of drive. Its wide use on motor trucks is due chiefly to this latter feature. In connection with passenger car drives its silence of operation is of more importance. Other characteristics which influence the selection of worm drives for particular services are that with an under-mounted worm (worm below wheel) the propeller shaft lies very low and permits of a low frame and low body floor, while with an over-mounted worm the road clearance at the center of the rear axle is greater than with bevel gear drive.

Up to about forty years ago the worm and gear were considered merely as a means of transmitting motion, as distinct from a means for transmitting power. As would be expected, with teeth inaccurately cut, worm and wheel crudely mounted, and no lubricant on the tooth contact surfaces, the efficiency was low, and wear was rapid. Moreover, the gear was not reversible; that is, power could not be transmitted from the gear to the worm. Worm gearing was first developed for power-transmission purposes in connection with electric motors. These were the first high-speed motors to come into practical use, and high reduction ratios were needed for many applications, such as that to elevators and hoists. Worm drive was first applied to motor vehicles in England, by F. W. Lanchester and the Dennis Brothers.

Advantages of Worm Drive—Several advantages of worm drive have been pointed out already. The chief field for this form of final drive is that of the heavy truck, which requires a reduction ratio of more than 6 to 1, which cannot be ob-

tained in a single step by any other form of final drive. The greater simplicity of a single reduction gear as compared with a double reduction one is an important advantage from the commercial point of view, as it permits of lower cost of manufacture. The efficiency of the worm drive is somewhat lower than that of a comparable single-reduction drive of the bevel gear or chain type, but somewhat higher than that of a double-reduction mechanism. This statement must be taken as applying to the respective types of transmission in a general way and not necessarily in any specific case, for much depends upon details of design. With worm drive, for instance, there are three distinct sources of loss, viz., tooth contact sliding friction, oil churning in the gear housing, and bearing friction, and each of these losses can be varied independently of the others.

Finally, the worm gear has the advantage that it renders it possible to provide a wide range of gear ratios without changing the distance between the axes of the worm and the wheel and without changes in the adjacent parts. The worm drive gives a symmetrical rear axle which is comparatively easy to assemble. A worm and wheel of the automotive type are shown in Fig. 1.

Materials for Worm and Wheel—The worm is invariably made of case-hardening steel. Where there is plenty of material below the teeth, carbon steel (No. 1015 or 1020) will do, but the nickel steels Nos. 2315 and 2320 are extensively used, as they give a stronger core. Case hardening is effected after the teeth are cut. This process will distort the teeth more or less, and they are therefore corrected by grinding, which can be readily done on special worm grinding machines. The wheel presents a decidedly more difficult material problem than does the worm. It might be thought that, since the teeth of worm and wheel are always separated by a film of oil, it would not make much difference of what material they are made, provided they have sufficient strength to withstand the tooth loads. This view, however, has proven to be erroneous, and the reason is believed to be that different metals have different capacities for maintaining the oil film. It is also possible that, especially with over-mounted worms, there is practically no oil on the teeth when the car is first being started after a long rest, and that some metals will withstand momentary operation without an oil film better than others.

The blanks for the worm wheels are always made of bronze, and three different bronzes are in use for the purpose. One of these is S. A. E. standard phosphor bronze for gears

(Specification No. 65), which is composed of 88–90 per cent copper, 10–12 per cent tin, 0.10 to 0.30 per cent phosphorus, and lead, zinc and other impurities not exceeding 0.50 per cent. Castings made of this alloy should show an ultimate strength of 35,000 lbs. per sq. in., an elastic limit of 20,000 lbs. per sq. in. and an elongation of 10 per cent in 2 in. or proportionate gage length. In order to increase the strength and hardness of the gears, the blanks are chill-cast, and when thus cast the above bronze will show a Brinell hardness of 75–90.

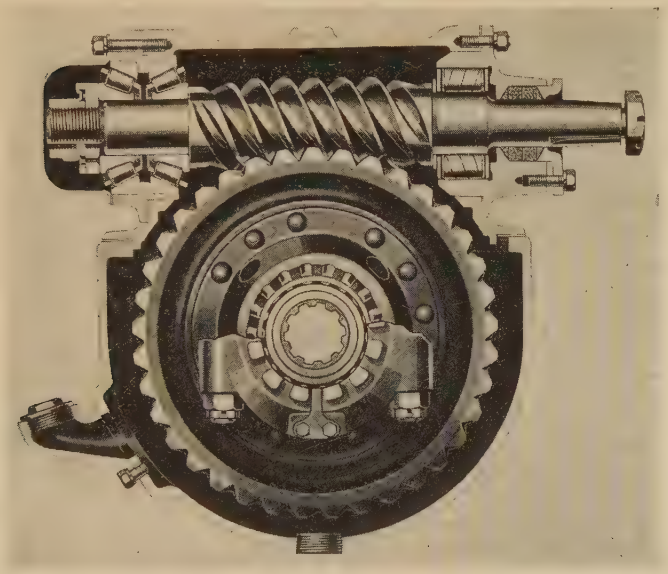


Fig. 1.—Automotive-Type Worm and Wheel (Over-Mounted).

The second material is aluminum bronze, composed of 89 per cent copper, 10 per cent aluminum and 1 per cent iron. This alloy is not chill-cast but is subjected to a heat treatment which imparts the following physical properties: Ultimate strength, 80,000–90,000 lbs. per sq. in.; elongation, 4–10 per cent, and Brinell hardness, 170–200. This aluminum bronze can be die-cast, but, unfortunately, in most applications of worm gearing the production is not sufficiently large to make this possibility commercially available.

The third material is a nickel bronze of the following approximate composition: Copper, 88.50 per cent; tin, 10 per

cent; nickel, 1 per cent; lead, 0.25 per cent and phosphorus, 0.25 per cent. This alloy is chill-cast, and in that condition shows an ultimate strength of more than 40,000 lbs. per sq. in., an elongation of 4 per cent and a Brinell hardness of 90–110.

Great efforts have been made to improve the physical properties of worm gear bronzes by improvement in foundry methods. The blanks are now usually cast with chills on three sides, and in England the centrifugal casting method has been applied to increase the density of the metal. In some cases the blanks are cast with the teeth practically formed in the mold, leaving only a small amount of metal to be removed in the gear-cutting process. This not only speeds the gear-cutting operation but retains more of the effect of the chills on the surfaces of the teeth.

Theory of Worm Gearing—The worm gear as applied in motor vehicle drives is similar to a helical gear, the worm being always of the multiple thread type, and some of the rules of helical gearing therefore also apply to worm gear. In a worm and worm wheel the gear reduction is equal to the quotient of the number of teeth in the worm wheel by the number of threads in the worm. The lead of the worm is the distance in the direction of the worm axis corresponding to one complete revolution of the worm thread. The angle of lead is the angle made by the pitch line of the worm thread with a plane perpendicular to the worm axis (also the angle made by a worm wheel pitch line with the worm wheel axis). In connection with helical gears it is customary to speak of the angle of spiral, which is the angle made by the pitch element of the gear tooth with the gear axis, and in case the two axes are at right angles to each other (as in a worm and wheel) the angles of spiral for the two gears together make a right angle. The angle of lead of the worm is equal to the angle of spiral for the worm wheel, while the complement of the angle of lead is the angle of spiral for the worm.

Following are definitions of some terms used in connection with worm gearing:

Circular pitch (of wheel)	= (Pitch diameter × 3.1416)/No. of teeth.
Axial pitch (of worm)	= Lead/No. of threads.
Circular pitch of wheel	= Axial pitch of worm.
Normal circular pitch	= Circular pitch × cos of angle of lead.
Normal circular pitch	= 3.1416/normal diametral pitch.

In calculating worms and worm wheels the following equations may be used:

WORM

Pitch diameter	= (No. of threads × axial pitch)/(3.1416 × tan of lead angle).
Lead	= Pitch diameter × 3.1416 × tan of angle of lead.
Addendum	= Normal circular pitch/3.1416.
Dedendum	= Normal circular pitch/2.716.
Outside diameter	= Pitch diameter + (2 × addendum).
Root diameter	= Pitch diameter − (2 × dedendum).

WHEEL

Pitch diameter	= (No. of teeth × circular pitch)/3.1416.
Lead	= (Pitch diameter × 3,1416)/tan of angle of lead.

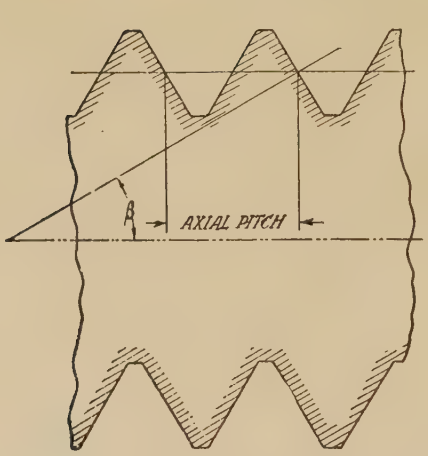
Addendum and dedendum, same as for worm.

Throat diameter = Pitch diameter + (2 × addendum).

The center distance or distance between the axes of worm and wheel is equal to one-half the sum of the two pitch diameters. Worm and wheel must be cut both either with right-hand threads or with left-hand threads. In an automobile drive with the engine rotating right-handedly as usual, worm and wheel must be cut with right-hand threads when the worm is placed on top of the wheel, and with left-hand threads when the worm is at the bottom.

If we cut a very thin section from the middle of the worm wheel we have a spur gear. If we cut a corresponding section from the worm, we have a rack, and since the flanks of a rack tooth to properly mesh with an involute gear must be straight lines, the faces of the worm teeth are straight.

Pressure Angle—In speaking of the inclination of the tooth contour, a distinction must be made between the normal pressure angle and the axial pressure angle. The axial pressure angle is the angle made by a line in a plane through the worm axis and perpendicular to the tooth flank, with the worm axis, and is represented by β in Fig. 2. This angle is evidently one-half of the included angle of the tooth in the section plane. The normal pressure angle a is the angle included by two lines in a plane cutting the tooth normally



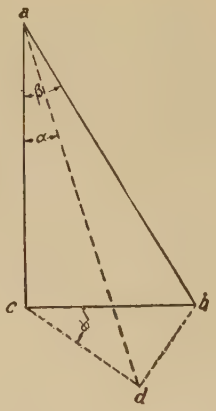


FIG. 2 (LEFT).—LONGITUDINAL SECTION THROUGH WORM WITH 30° AXIAL PRESSURE ANGLE.

FIG. 3 (RIGHT).—RELATION BETWEEN AXIAL AND NORMAL PRESSURE ANGLES.

or at right angles to its elements, these lines both passing through the pitch point in the tooth contour, one being tangent to the tooth contour at that point and the other passing through the worm axis. The relation between the axial pressure angle and the normal pressure angle is illustrated in Fig. 3. In this figure the line cd is supposed to be perpendicular to ac and not in the plane of the paper.

$$\tan \beta = \frac{bc}{ac}$$

$$\tan \alpha = \frac{cd}{ac}$$

$$cd = cb \cos \phi.$$

Substituting this value of cd in the preceding equation we have

$$\tan \alpha = \frac{cb \cos \phi}{ac} = \tan \beta \cos \phi.$$

That is, the tangent of the normal pressure angle is equal to the tangent of the axial pressure angle multiplied by the cosine of the lead angle.

Most makers of worm gears for automobile transmission

use an axial pressure angle of 30°. With a lead angle of 35° this corresponds to a normal pressure angle of 25° 19′. Normal pressure angles of 22½° and 14½° have been used, but with these there is undercutting of the wheel teeth if the number of teeth is small. With the Hindley type of worm there is further difficulty that the worm could not be assembled with the wheel if the pressure angle were too small.

Center Distance—The distance between the axis of the worm and the axis of the wheel bears a close relation to the maximum torque that can be transmitted, and therefore to the total weight of the vehicle. In motor truck practice the center-to-center distance found in ½-ton and 1-ton trucks, is about 6¾ inches. For 5-ton trucks a center-to-center distance of about 9½ inches is used, and for worm gears for motor trucks of other capacities the center distance may be found approximately by the following equation:

$$L = 0.7\,t + 6 \text{ in.}$$

where t is the truck capacity in tons.

As the worm pitch diameter

$$d = \frac{np}{\pi \tan \phi}$$

and the wheel pitch diameter

$$D = \frac{Np}{\pi};$$

and as the center distance

$$L = \frac{D + d}{2},$$

we have

$$L = \frac{\frac{np}{\pi \tan \phi} + \frac{Np}{\pi}}{2} = \frac{p}{2\pi}\left(\frac{n}{\tan \phi} + N\right)$$

Capacity of Worm Gears—The question of the amount of power which a given worm will transmit is a very involved one. It depends more upon the capacity of the gear for disposing of the heat than upon the mechanical strength of its teeth. As the temperature of the worm and wheel and of the oil bath rises, the oil becomes thinner, and if it should become too thin it would be squeezed out from between the teeth, and cutting would ensue. To prevent the oil from getting too hot and too thin, provisions must be made for

dissipating the heat from the differential housing as fast as it is generated while the temperature of the housing and its contents is still tolerable.

The limiting load capacity of a worm and gear also depends upon the angle subtended by the wheel teeth at the axis of the worm or the degree to which the wheel teeth envelop the worm. Experiments have shown (what would have been expected) that if the face width of the gear is successively reduced, the drive will fail at lower and lower loads. However, the wheels are now generally so made as to give the largest possible area of tooth contact. According to David Brown & Sons, Ltd., British worm gear specialists, while the face width should be as great as possible within reason, the throat should not envelop the root of the worm by an amount greater than 4.71 $\times$ the normal circular pitch, in order to prevent cutting the wheel teeth to an edge. In American practice the sides of the worm wheel rim are generally chamfered along lines radial to the worm axis, but according to the above-mentioned firm this is unnecessary, and the corners of the rim preferably should be turned square, with a small radius, as shown in Fig. 5, as this makes the machining easier and gives a greater effective tooth area.

Proportions for Passenger Car Drive Gears—For the same maximum engine power to be transmitted, worm gear final drives for high-speed passenger cars are made much smaller than those for trucks and other heavy duty vehicles. This is permissible first, because in a passenger car the average load factor is generally much smaller and, second, because the average vehicle speed is much higher and hence the differential housing can get rid of the heat generated in it at a more rapid rate.

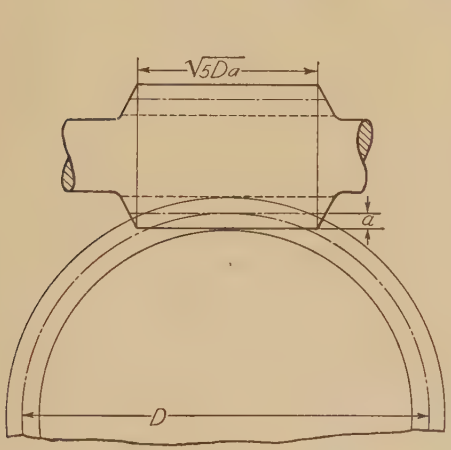
What limits the capacity of a worm drive is the rise of the temperature of the lubricating oil, which latter is dependent upon the capacity of the differential housing to dissipate heat. This ability to dissipate heat is substantially proportional to the surface area of the housing, which is proportional to the square of its linear dimensions. Hence one might expect the capacity of a worm gear drive to increase as the square of its linear dimensions. Actually the capacity varies somewhat more rapidly, because in a passenger car, as a rule, the higher the engine power the lower the average load factor of the engine and the higher the average car speed and, therefore, the greater the cooling facilities of the differential housing. Data at hand indicates that the horse power transmitting capacity of worm gears in passenger car drives varies as the 2.4th power of the linear

dimensions and, inversely, the linear dimensions of such a gear must be made proportional to the 0.42th power of the horse power to be transmitted. A usable value for the required center distance is given by the equation

$$L = 0.75\, P^{0.42} \text{ in.},$$

where P is the maximum engine horse power. This gives a center distance of 5¼ in. for a 100-hp. engine and of 3⅞ in. for a 50-hp. engine.

In automotive worm gearing the axial pitch system is used; that is to say, the axial pitch, rather than the normal diametral pitch, is given a value in round figures. Lead angles are generally held between 30° and 40°, this being the range of highest efficiency. For large reductions, such as 14 to 1 or



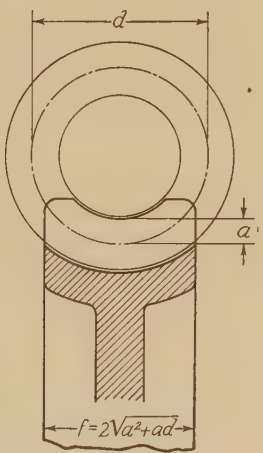


Fig. 4 (left).—Diagram for Length of Worm.
Fig. 5. (right).—Diagram for Width of Wheel Face.

15 to 1, as sometimes used in heavy trucks, the worm is made with only two leads, while for smaller reduction ratios it is made with four and even with six leads. The number of teeth in the worm wheel is generally kept within the limits of 20 and 30. The procedure in calculating a worm gear drive is then somewhat as follows:

Calculation of Gear Elements—We first determine the approximate center distance required by one or the other of the two formulae given, according to whether the drive is for a truck or for a high-speed vehicle. Next we select suitable

numbers of leads and teeth to give the desired reduction ratio. After having also selected the lead angle we insert in the formula for center distance, viz.,

$$L = \frac{p}{2\pi}\left(\frac{n}{\tan \phi} + N\right),$$

all of the known values and solve for p, the axial pitch. We then select for the axial pitch the dimension in full sixteenths or full eighths of an inch nearest to the calculated value, and finally, by inserting this value in the above equation, calculate the actual center distance.

From the production standpoint there is a certain advantage in using a single "standard" lead angle and axial pitches in definite steps, as this will reduce the amount of equipment required where worm gears for many different applications are out. However, with production on a very large scale this advantage is of little importance, and, besides, if it is desired to mount gear sets of different ratios interchangeably on the same centers, it is necessary to change either the axial pitch, the lead angle, or both.

Length of Worm and Face of Wheel—It is not desirable to make the worm so long and the wheel face so wide that these parts extend to the very limit of possible tooth contact. A more practical plan is to proportion worm and wheel as indicated in Figs. 4 and 5. This gives for the length of the worm

$$l = \sqrt{5Da},$$

and for the face of the wheel,

$$f = 2\sqrt{a^2 + ad},$$

where a is the addendum; d, the worm pitch diameter and D, the wheel pitch diameter. These rules apply to straight worm sets only.

Theoretical Efficiency—Most of the older mechanical handbooks and textbooks on worm gearing give a formula for the efficiency of such gearing, expressed in terms of the lead angle, the pressure angle and the coefficient of friction. One much-quoted formula is as follows:

$$\varepsilon = \frac{\cos \alpha - f \tan \theta}{\cos \alpha + f \cot \theta}.$$

This indicates that if the friction coefficient does not exceed 0.02 the efficiency is nearly constant for lead angles varying between 20° and 70° and falls off to zero as lead angles of

zero and 90° are approached. Unfortunately this equation is based on the assumption that the friction is proportional to the load on the contact surface, which is not the case when the surface is well lubricated, as it always should be in an automotive-type worm gear.

Thrust and Radial Bearing Loads—The thrust load on the worm is equal to the tangential force on the wheel pitch circle and the thrust load on the wheel is equal to the tangential force on the worm pitch circle. The effect of tooth friction may be neglected. Denoting the full load torque of the engine by T, the worm pitch diameter by d, the wheel pitch diameter by D and the reduction ratio by r, we have

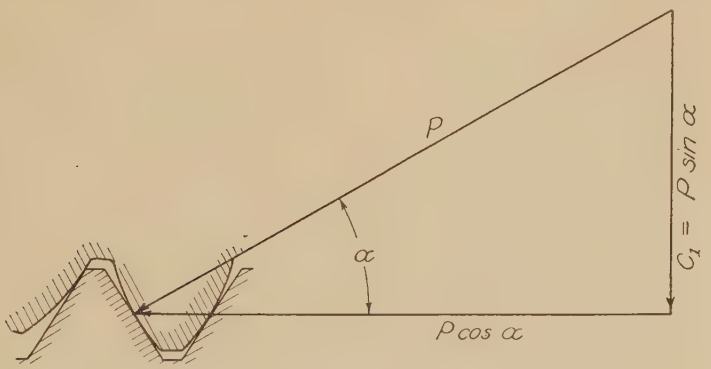


Fig. 6.—Components of Normal Tooth Pressure.

for the tangential force on the worm pitch circle, and hence for the thrust load on the wheel, at full engine load and direct drive,

$$L_t = \frac{24T}{d},$$

and for the tangential force on the wheel pitch circle; and, consequently, the thrust load on the worm,

$$l_t = \frac{24Tr}{D}.$$

The radial loads on both the worm and the wheel shafts are made up of two components which act at right angles to each other. The first is due to the pressure angle of the teeth; it passes through the center of tooth contact and is perpendicular to both the worm axis and the wheel axis.

This is the force tending to separate the shafts and is, of course, the same for both the worm and the wheel. If we denote the normal tooth pressure by P, then this component C_1 is equal to $P \sin \propto$ (Fig. 6). But, as may be seen from Fig. 7,

$$P = \frac{l_t}{\cos a \cos \phi} = \frac{L_t}{\cos a \sin \phi}.$$

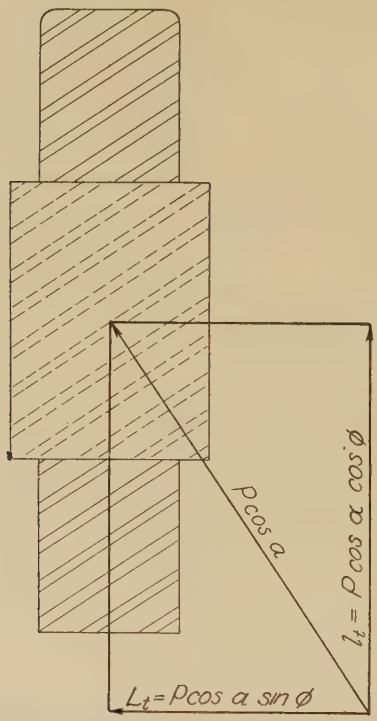


FIG. 7.—DIAGRAM SHOWING THE TANGENTIAL FORCES ON WORM AND WHEEL, WHICH ARE EQUAL, RESPECTIVELY, TO THE THRUST LOADS ON THE WHEEL AND WORM SHAFTS.

Hence

$$C_1 = P \sin a = \frac{l_t \tan a}{\cos \phi} = \frac{L_t \tan a}{\sin \phi}.$$

The other component of the radial load, C_2, is different for the worm and the wheel, respectively. For the wheel it is equal to the thrust load on the worm, l_t, and for the worm it is equal to the thrust load on the wheel, L_t. That the two

components of the radial load on each shaft are at right angles to each other may easily be shown. Take, for instance, the components of the radial load on the wheel shaft. The first component, C_1 is perpendicular to the worm shaft, while the second component, the thrust load on the worm shaft, naturally is parallel to that shaft and hence must be perpendicular to the first component. Therefore, the total radial load on the wheel shaft is equal to the square root of the sum of the squares of the components

$$L_r = \sqrt{l_t^2 + \left(l_t \frac{\tan a}{\cos \phi}\right)^2} = l_t \sqrt{1 + \left(\frac{\tan a}{\cos \phi}\right)^2}$$

But

$$\frac{\tan a}{\cos \phi} = \tan \beta,$$

so that

$$L_r = l_t \sqrt{1 + \tan^2 \beta}$$

and if β, the axial pressure angle, has a constant value of 30° then

$$L_r = 1.155 l_t.$$

That is, the radial load on the wheel bearings is 15.5 per cent greater than the thrust load on the worm bearings.

Similarly, the total radial load on the worm shaft is

$$l_r = \sqrt{L_t^2 + \left(L_t \frac{\tan a}{\sin \phi}\right)^2} = L_t \sqrt{1 + \left(\frac{\tan a}{\sin \phi}\right)^2}$$

which, after the value of tan a is substituted, becomes

$$l_r = L_t \sqrt{1 + \left(\frac{\tan \beta}{\tan \phi}\right)^2}.$$

The normal pressure between the teeth of worm and wheel is given in terms of known factors by the following equation:

$$P = \frac{24T}{D \cos \alpha \cos \phi} \text{ lbs.},$$

and the rubbing speed at the tooth contact surface by

$$v = \frac{n \pi d}{12 \cos \phi} \text{ ft. p.m.}$$

where T is the worm wheel torque (worm torque multiplied by reduction ratio); D, the wheel pitch diameter in inches; α, the normal pressure angle; ϕ, the worm lead angle; n, the

angular speed of the worm in revolutions per minute; and *d*, the worm pitch diameter in inches.

Hindley Worm Gears—All worm wheels are throated, that is, the face of the gears, instead of being a cylindrical surface, is machined to a radius slightly larger than the root radius of the worm. This "throating" operation is generally performed simultaneously with the cutting of the teeth, the throat being formed by the hob. The object, of course, is to increase the tooth contact area over that obtainable with an unthroated face. It is also possible to "throat" the worm, and this form of worm is known as the Hindley or "hour-glass" type. Such a worm ensures increased bearing surface if correctly mounted. However, its machining involves some difficulty and it requires additional care in mounting. The ordinary straight worm must be mounted

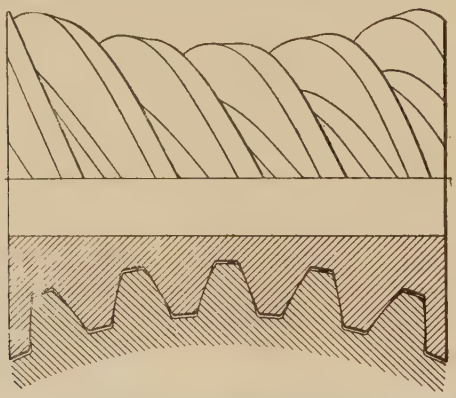


FIG. 8.—HINDLEY TYPE OF WORM (Shown Longer than Made in Practice, to Bring Out the Hour-Glass-Effect More Clearly).

accurately in two planes; that is, the worm axis must be at a definite distance from the wheel axis, and it must also be in the median plane of the worm wheel. The Hindley or "hour-glass" worm, on the other hand, must be accurately located in three planes; that is, the worm axis must be a definite distance from the wheel axis; it must be in the median plane of the worm wheel, and the median plane of the worm must include the wheel axis. In cutting a Hindley worm the cutting tool must be mounted so as to turn around a center at a distance from the cutting edge equal to the radius of the worm wheel, and it must be fed in the direction perpendicular to the plane of rotation of the worm. The teeth

of Hindley worms cannot be corrected by grinding after hardening, but must be cleaned or finished by hand, which is especially objectionable in American practice.

The majority of worm drives in England seem to employ straight worms, but the Lanchester worm, which is used on several foreign makes of cars and has been introduced in this country, is of the Hindley type. When proper manufacturing equipment is available the manufacturing difficulties vanish, and there remains only the greater difficulty involved in properly mounting the Hindley worm to balance the advantage of a lower unit tooth pressure. The Hindley worm is made shorter than the straight worm, usually between one-fifth and one-quarter the wheel diameter. This is necessary because otherwise it would be impossible to mesh the worm

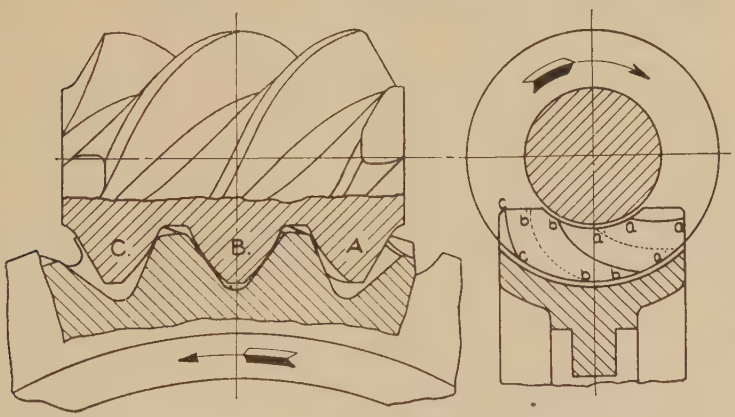


Fig. 9.—F. J.-Type of Worm Gear.

and wheel. Fig. 8 illustrates the principle of the Hindley worm.

FJ Form of Worm Tooth—If an axial section is taken through a conventional automotive worm, a rack is obtained whose straight tooth sides form an included angle of 60°. Practically all theoretical investigations of worm tooth contact have been confined to the center plane, but it will be realized that, since contact extends over the whole width of the gear, other planes are of equal or at least nearly equal importance.

As a result of a study of tooth contact in other planes than the central one, a new form of worm tooth was developed in England by F. J. Bostock in 1920 and shortly after-

ward adopted for automotive worm gearing by the firm of David Brown & Sons, Ltd. Whereas the conventional worm is generated with a cutter having straight cutting edges which intersect the worm axis, the FJ worm is generated by means of a cutter whose straight edges are tangent to a cylinder coaxial with the worm and which has a diameter substantially equal to the root diameter of the worm. An axial section through an FJ worm shows it to have teeth with slightly convex sides, while the teeth of the gear in the same plane tend to be concave.

A worm and gear with the FJ form of teeth are illustrated in Fig. 9. The view on the left shows the forms of

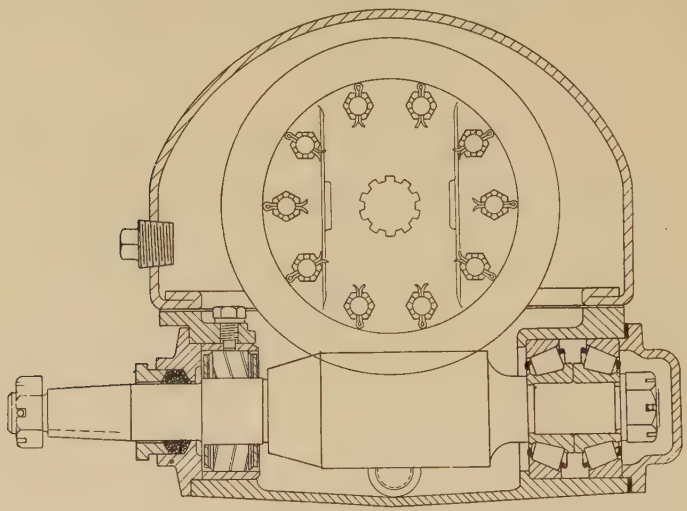


FIG. 10.—WORM DRIVING GEAR OF PASSENGER-CAR AXLE, LONGITUDINAL SECTION.

the teeth in a plane through the worm axis and perpendicular to the gear axis, while the view on the right shows the gradual progress of the line of contact on the gear tooth from the time the worm thread first contacts with it up to the time tooth contact is broken. With the conventional worm and gear the line of contact is at all times more or less parallel to the gear axis, and it progresses from the top toward the root of the gear tooth. Advantages claimed for the FJ form of worm tooth are that the whole of the tooth surface is utilized more nearly uniformly, which equalizes tooth wear, and that the line of contact always makes a considerable

angle with the direction of the sliding motion, which tends to maintain the film of oil between the teeth.

Mounting of Worm Gears—Practically all automotive worms are made integral with their shafts. This in most cases permits of making the pitch diameter of the worm smaller, which makes for increased efficiency. In American practice the worms of passenger car worm drives are always undermounted, because this gives a lower propeller shaft and permits of lowering the car frame and body, while the worms of truck worm drives are always over-mounted, as this gives a greater road clearance at the center of the rear axle. Tooth contact conditions are better with the under-mounted worm, as the contacting surfaces are always submerged in oil, hence

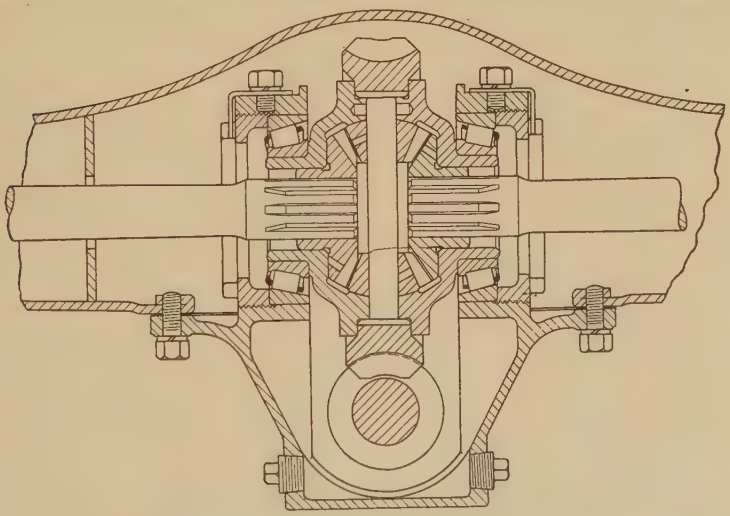


FIG. 11.—WORM DRIVING GEAR OF PASSENGER-CAR AXLE. TRANSVERSE SECTION.

there is greater assurance that the film between them will be maintained. With an under-mounted worm the engine and transmission must be set in the frame at a somewhat greater angle in order to obtain a straight-line drive, and it is more difficult to prevent oil leakage around the pinion shaft, which is located below the oil level in the differential housing. However, the use of a stuffing box on the worm shaft seems to solve this problem in a satisfactory way.

Worm driven axles are now always made of the differential-carrier type. All of the heavier worm-driven axles

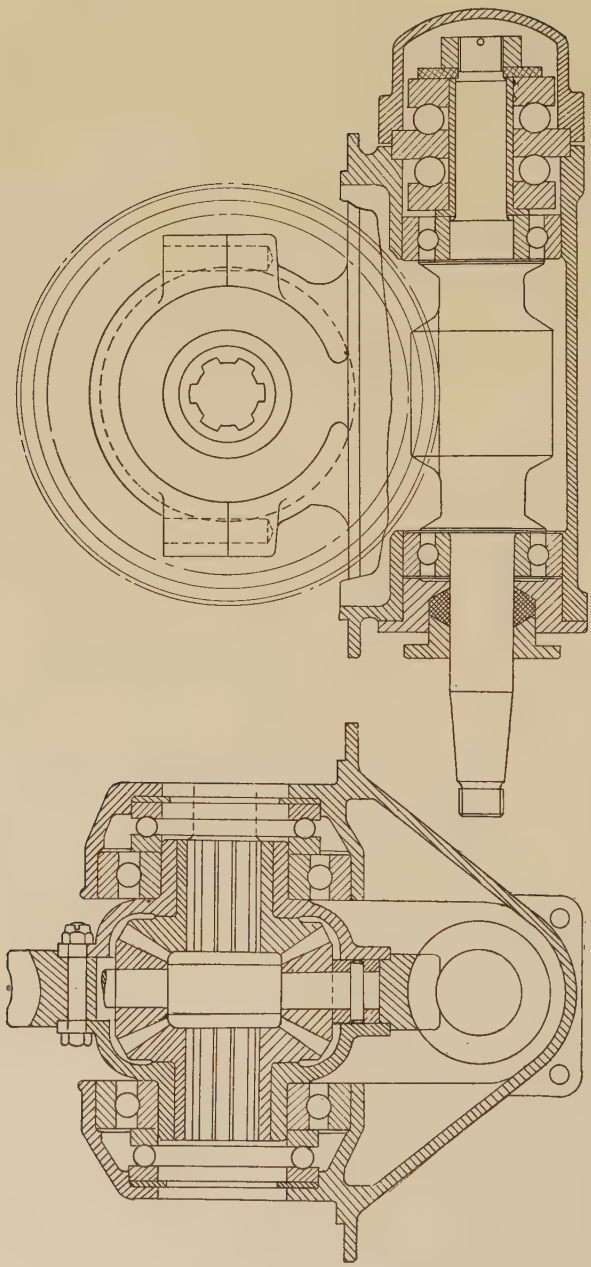


FIG. 12.—BRITISH UNDER-MOUNTED TYPE OF WORM DRIVE.

have housings of either cast steel or malleable iron, while the housings of lighter axles are made of pressed steel. The central portion of the housing is made of bulbous form with a large opening in the top or bottom, according to whether the worm is over- or under-mounted, through which the differential carrier may be inserted.

Figs. 10 and 11 show a typical worm drive for passenger cars. The worm wheel is carried between the halves of the differential housing, ten bolts with castellated nuts passing through flanges on the housing and lugs on the inside of the worm gear ring. The differential gear is mounted on taper roller bearings on both sides and can be adjusted axially to bring it centrally over the worm. The worm is integral with its shaft, which latter is mounted on two taper roller bearings at its rear end, which take thrust loads in both directions, and in a flexible cylindrical roller bearing at its forward end. Attempts have been made in the past to mount worm shafts on a taper roller bearing at each end, each bearing taking thrust in one direction, but this has generally led to trouble, owing to the very considerable heat expansion of the worm shaft when full power is being transmitted continuously. The bearing mounting referred to tends to prevent such expansion and generally leads to buckling of the shaft and to consequent faulty tooth action. With the worm mounting shown in Fig. 10 the worm is entirely free to expand and contract. A feature of the design to which attention may be called is that the shaft diameter is considerably increased immediately inside the bearings, so the deflection of the shaft will be minimized.

Figure 12 shows an under-mounted worm drive of British design. Practically the only differences as compared with the design previously shown are in the use of ball bearings all around and in a somewhat different proportioning of the worm and wheel sections. It will be noticed that two radial bearings and two thrust bearings are used for the worm and wheel each. When American engineers use ball bearings for mounting the worm they usually fit a double-row angular contact bearing at one end to take thrust in both directions, and a single row annular bearing at the other, with its outer race free to slide axially. Double row bearings also are favored for mounting the differential gear of worm drive axles if ball bearings are used.

Figure 13 shows a worm-driven axle designed for a gas-electric bus with double motor drive. The axle has no differential gear but two separate worm gears, each in its own housing, the two housings being separated by an I-section placed on its side, as it were. This design shows two features

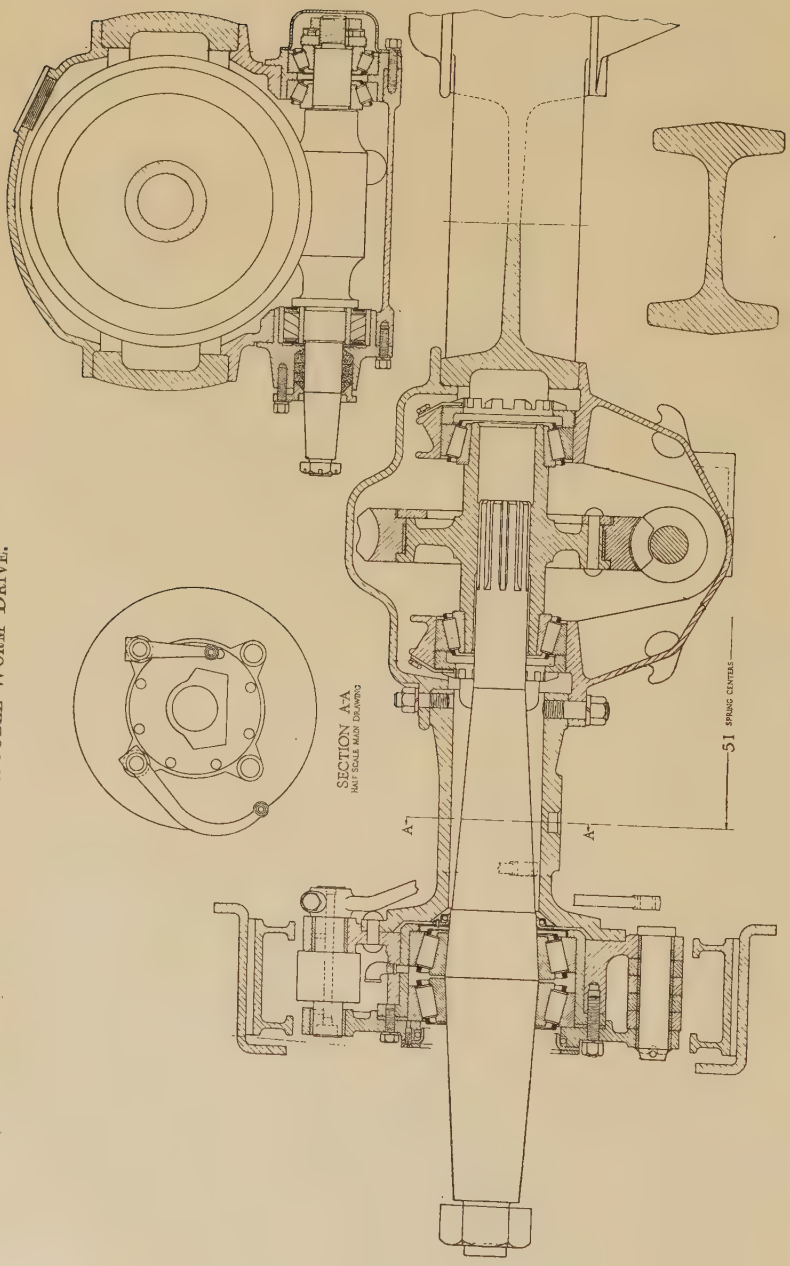


FIG. 13.—DETAILS OF BUS AXLE WITH DOUBLE-WORM DRIVE.

quite popular in heavy drive axles. One of these is the method of fastening the ring gear to its center web or drum, the equivalent of the differential housing. A portion of the width of the inner face of the gear ring is splined or key-seated, while the rest is left full. Thus the wheel is both accurately piloted and drives positively. The other feature referred to is in the outboard bearings of the axle. These, it will be seen, consist of double roller bearings of the tapered type, the outer races of which are held rigidly in the axle housing, so that the bearings take end thrust in both directions and this thrust does not have to be transmitted through the axle shaft from one side of the chassis to the other. Usually there is a shallow flange on the axle shaft between the inner races of the inner bearings, which is absent in this design, in which both inner races are mounted merely on tapered seats.

Worm-driven Bus Axle—A typical worm-driven axle as used in Daimler buses in service in London is illustrated in Figs. 14 and 15. These buses, when carrying a full complement of passengers, weigh 13,440 lbs., of which 8960 lbs. is on the rear axle. They are fitted with four-cylinder Daimler-Knight engines of 4.6-in. bore and 4.4-in. stroke. The worm, which is of the Lanchester hour-glass type, has four leads, while the wheel has 29 teeth, making the reduction ratio 7¼ to 1. The center distance between worm and wheel is 7¾ in.

It will be noticed that the axle is of the full-floating type, the cast steel wheels running on the outside of the axle tube on cylindrical roller bearings. Connection between the axle driving shafts and the wheel hub is made by bolted-on caps. The tread of the rear wheels is 69 in. and the distance between spring centers 50 in. The rear axle shafts, which are made of high tensile nickel steel, have an effective diameter of 1¾ in. Both ends of the driving axles, where they fit into the driving couplings, are upset.

The worm and wheel are carried by a removable gear carrier with its own radial and thrust bearings. The whole differential can be removed through the top opening after the axle shafts have first been withdrawn and to this end it is not necessary to first remove the bearing caps which are held by long through bolts. The differential is of the two-pinion type. Brake support and spring seat are in a single casting. A notable feature is the large capacity of the central housing. There is no stuffing box inside the axle tubes, so the lubricant can work right out to the hub bearings, but a packing is provided at the inner end of the wheel hub to prevent oil working onto the brake surfaces.

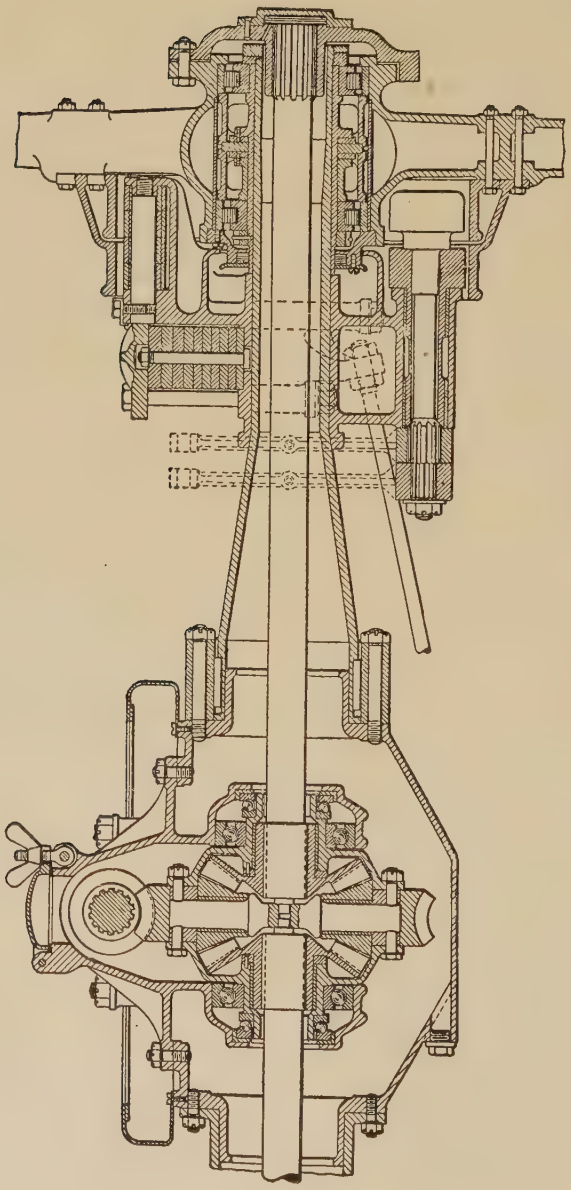


Fig. 14.—Section through Daimler Worm-Driven Bus Axle.

Efficiency of Worm Gears—In 1912 several series of tests of Lanchester worm gears were made by the National Physical Laboratory of England for the Daimler Motor Company on a special testing machine designed by Mr. Lanchester (Fig. 16). This machine is based on a principle similar to that of the electric cradle dynanometer. It has been repeatedly pointed out that the rear axle housing tends to turn in the direction opposite to that of the axle shafts, with a torque exactly equal to that of the axle shafts. Therefore, by mounting the rear axle housing in ball bearing supports and holding it from rotation by means of a weight on an arm secured to it, we have a measure of the rear axle torque. If the axle is worm driven, the housing also has a tendency to turn in a plane

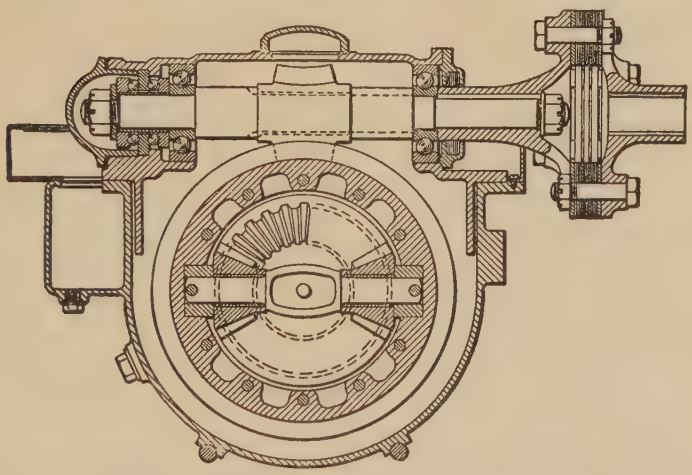


Fig. 15.—Section through Worm Gear of Daimler Bus Axle.

perpendicular to the worm axis in the direction opposite to that of the worm shaft, with a torque equal to that of the worm shaft. Lanchester, therefore, gives his worm gear housing such a support that it may rock in two vertical planes at right angles to each other. He then measures the torque on the propeller shaft and on the axle shaft, respectively, by balancing the housing in both planes. This he does by means of a single weight suspended from a knife edge parallel with and at a given distance from the axle shaft axis.

If we denote the torque on the worm shaft by t, that on

the axle shafts by T and the worm gear reduction ratio by r, then if there were no loss in the gear we would have

$$tr = T$$

As a matter of fact T is always less than tr, and the efficiency is measured by the ratio T/tr.

The testing machine comprises a cradle consisting of two wheels coupled by bridges. The cradle is supported by four ball-bearing rollers and power is transmitted to the worm and from the rear hub universal jointed shafts, the joints being of the ball-bearing type. A balance arm is fixed to one side of the case parallel to the worm shaft. At the end of this arm there is a transverse knife edge arm on which a

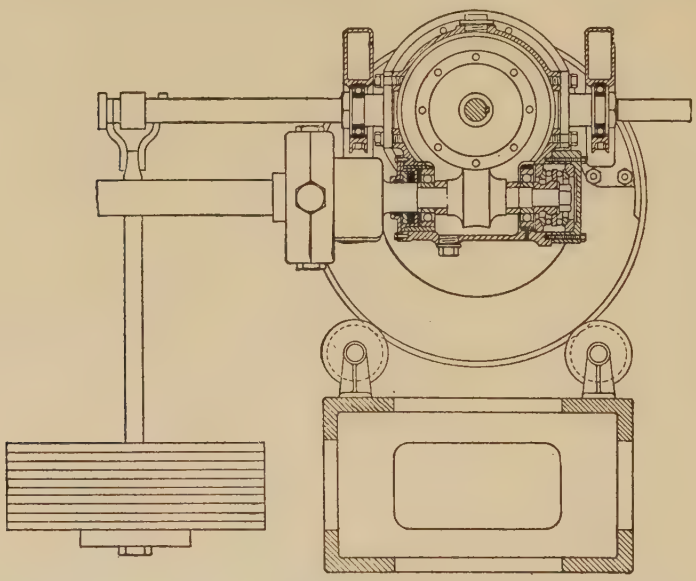


FIG. 16.—LANCHESTER TESTING MACHINE FOR WORM GEAR.

weight is suspended by means of a rod. The weight can be slid along the transverse arm by means of a finger wheel, and its distance from the axis of suspension can be read off on a dial.

The gear box is supported from the cradle on ball bearings in such a way that the axis of the worm intersects the axis of the cradle wheels. When the worm gear housing is in equilibrium the contact point of the knife edge is located

Table I—Worm Gear Test Results

Worm Speed, R.P.M.	Calculated H.P.	Efficiency, P.C.	Torque on Driven Shaft, Lb.-Ft.	Pressure on Thread, Lbs.
		8:33 Worm Gear		
1542	31.9	95.4	449	1205
383	7.9	93.9	449	1205
1532	45.6	95.7	645	1733
408	12.1	94.0	645	1733
1532	59.4	95.8	841	2258
403	15.6	93.8	841	2258
1532	73.3	95.7	1035	2786
373	17.8	93.7	1037	2786
1497	17.5	93.5	254	682
418	4.9	93.6	254	682
		8:35 Worm Gear		
1532	29.8	95.7	447	1200
418	8.1	94.6	447	1200
1512	42.3	96.2	643	1727
413	11.6	95.1	643	1727
1532	69.0	95.6	1035	2780
398	17.9	93.1	1035	2780
		9:34 Worm Gear		
1532	34.5	96.0	447	1200
393	8.9	95.2	447	1200
1527	49.5	96.6	613	1727
403	13.1	95.0	613	1727
1527	79.6	96.6	1035	2780
403	21.0	95.0	1035	2780

in the plane of the two axes of rotation. In operation the finger wheel is adjusted until the gearbox is in equilibrium. Then, as the same weight is used to measure the torque around each axis of support, the torques are proportional to the distances of the point of knife edge contact from the two axes of support, respectively. We found that

$$e = \frac{T}{tr} = \frac{T}{t} \times \frac{1}{r},$$

and since

$$\frac{T}{t} = \frac{OA}{AB} \text{ (see Fig. 17)}$$

$$e = \frac{OA}{AB} \times \frac{1}{r}.$$

In order to be able to make efficiency tests of large worm gears with a small expenditure of energy, Lanchester connects his driven shaft (or axle shaft), through a step-up bevel gear set and a belt to the worm shaft, the step-up ratio being slightly greater than the reduction of the worm and worm wheel, so that the belt always slips slightly. As a result only the power lost in the worm gear, bevel gear and in belt slip needs to be supplied from an outside source. The belt tension is adjusted until the weight hung from the knife edge is lifted and when midway between stops the arm is locked in position. Readings are then taken, and afterwards

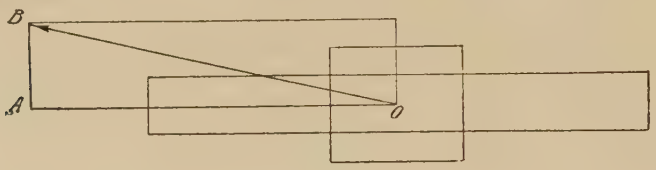


FIG. 17.—DIAGRAM OF TORQUE BALANCE.

the arm is released to see whether the torque has changed. Slight changes in torque do not affect the efficiency, consequently it is not necessary to constantly adjust the weight.

The chief results of the National Physical Laboratory tests on Lancaster worm gears are summarized on Table I on page 243, only the readings and calculated results for the highest and the lowest speed in each series being given.

CHAPTER IX

Double-Reduction Axles

It is customary in the industry to limit the term "double-reduction axle" to live axles in which a double reduction in rotative speed is effected by one pair of bevel and one pair of spur or herringbone gears, both enclosed in a housing at the center. However, the term may properly be applied to a much wider variety of designs, including all axles in which there is a double reduction of speed, whatever the type of gearing through which it is effected, and wherever that gearing may be located. In this chapter we will use the term "double-reduction axle" in this wider sense.

Owing to the fact that the propeller shaft extends at right angles to the rear axle, one reduction is always by bevel gears. The other pair may consist of spur gears, herringbone gears, or a spur pinion and an internal gear; and the second pair of gears may be located at the center of the axle, or there may be two pairs of second-reduction gears which may be located at the center or the ends of the axle, or in any intermediate position.

As a rule, the first reduction is by bevel gears, because the bevel gears create end thrust, and if they are used for the first reduction the end thrust will be smaller and therefore more readily taken care of. However, the opposite arrangement, in which the second reduction is by bevel gears, is also used, probably for manufacturing reasons.

Bevel-spur-gear drives are used to a large extent with axle housings of the banjo type, either cast or drop-forged. A typical design (Eaton) is shown in Fig. 1. The axle housing is cast and all of the gears are supported by a gear carrier which is bolted over a circular opening in the front of the housing. All shafts are in the same horizontal or slightly inclined plane, which is a rather uncommon arrangement. The axle shown is for trucks of the heaviest types, the load limit on its spring pads being 20,000 lbs. All bearings are of the double-row-ball type. The pinion shaft is mounted in

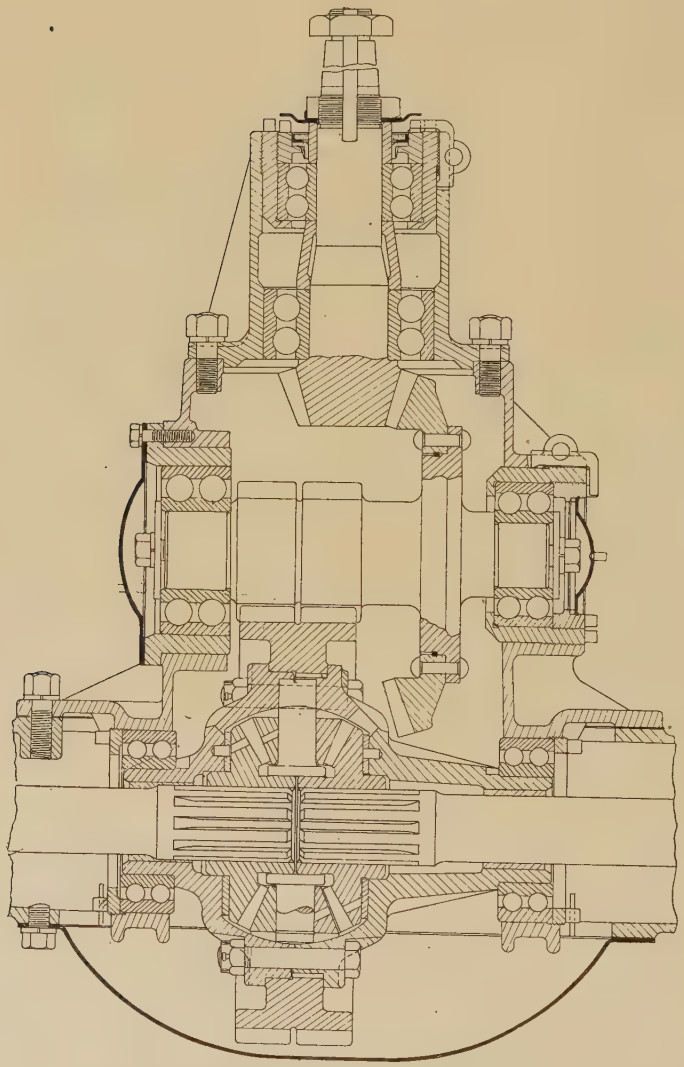


FIG. 1.—SECTION THROUGH CENTER HOUSING OF DOUBLE-REDUCTION AXLE WITH BEVEL AND HERRINGBONE GEARS.

two such bearings of which the forward one takes thrust in addition to the radial load. Thrust on the intermediate shaft is taken on the right-hand bearing. The bearing on the left carries considerably more radial load than that on the right, owing to the proximity of the herringbone pinion, whose pitch diameter is less than half that of the bevel gear, so that its tooth load and the radial bearing load resulting therefrom are greater than the corresponding loads due to the bevel gear. The bearings on the differential are adjustable axially, which is required in order to assure accurate meshing of the herringbone gears. In most double reduction

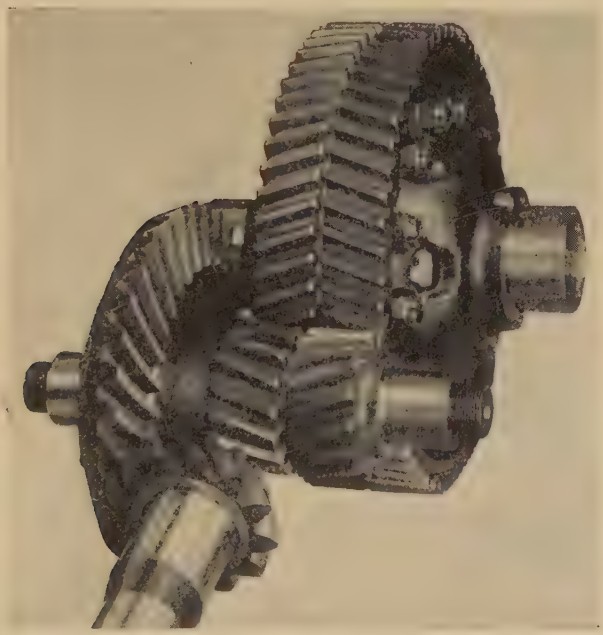


FIG. 1-A.—ASSEMBLY OF GEARS USED IN THE AXLE SHOWN IN FIG 1.

axles in which all the gears are located in a center housing the second reduction is by spur gears, but some manufacturers prefer herringbone gears because they operate more quietly and, unlike helical gears, they produce no end thrust. Their operation also is smoother, on account of the gradual rolling into contact of their teeth, which obviates shock.

A problem in connection with such gear trains is to design them so that they can be readily assembled. It will be seen

that the intermediate shaft is longer than the distance between its bearing bosses, and the shaft with the two gears on it must be "wiggled" into place before the bearings are inserted. Both bearings are supported in large bushings, and the bores of both bosses are quite large. The right-hand bearing on the intermediate shaft is made adjustable axially, so as to make it possible to ensure proper mesh of the bevel gears.

A side view and sectional views in different horizontal planes of the Mack bevel-spur-gear drive are shown in Figs. 2 and 3, respectively. The drop-forged axle is arranged so its center ring makes an angle of 45° with the horizontal, and all of the gears, including the differential, are supported

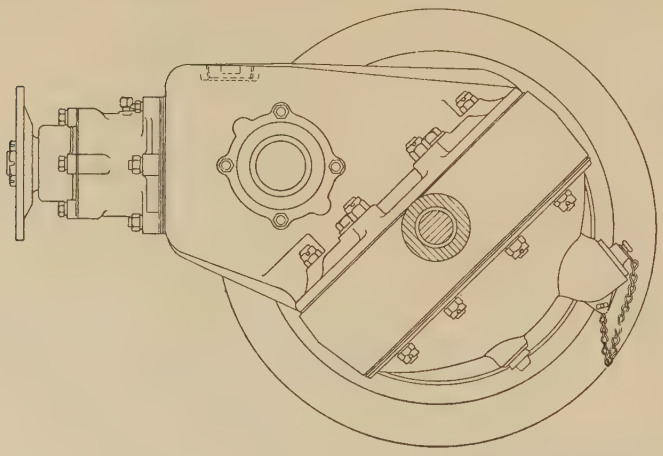


FIG. 2.—SIDE VIEW OF CENTER HOUSING OF DOUBLE-REDUCTION (BEVEL-SPUR) AXLE.

by a carrier which is bolted to the forwardly and upwardly facing side of the ring. The bevel gear is bolted to an integral flange on the intermediate shaft, and the spur pinion is secured to a slightly tapered seat on this shaft, being formed with teeth or keys at its end which engage into key-slots in an enlarged section of the shaft. This latter arrangement obviates weakening the pinion by cutting a keyway in it. At one side of the differential carrier there is an opening large enough to permit of introducing the intermediate shaft with both gears assembled upon it.

Spur-Bevel-Gear Drive—A drive in which the reduction by spur gears precedes that by bevel gears is illustrated in

Fig. 4. This has been used for many years by the White Company. The bearing mounting looks somewhat simpler than where the bevel reduction precedes the spur reduction, and all gears are straddle-mounted, which reduces bearing loads as compared with an overhanging mounting. The casting forming the gear carrier in this case serves also as the

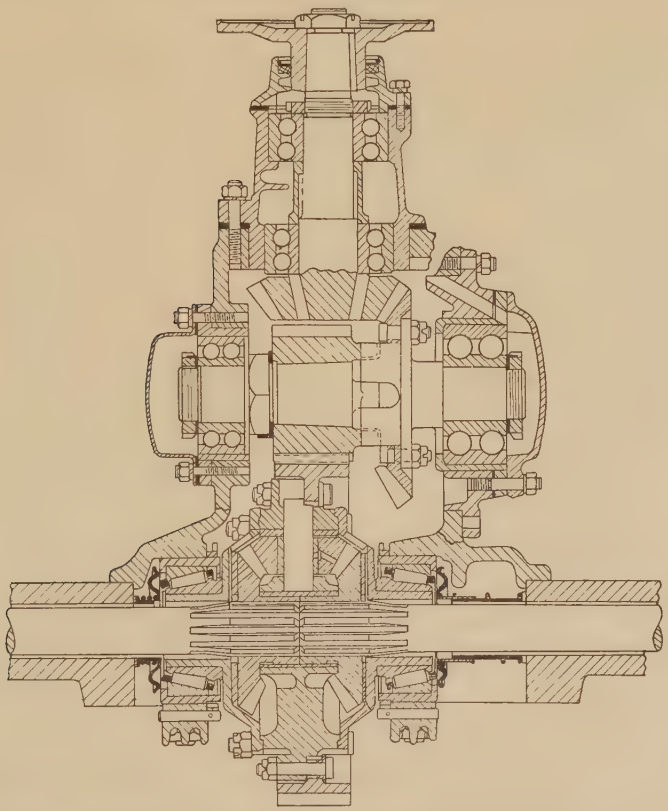


Fig. 3.—Sectional View of Drive Shown in Side View in Fig. 2.

center housing. It has a large opening at the rear through which the differential and bevel gear assembly can be introduced, and trumpet-shaped axle tubes are bolted to its sides.

Numerous different arrangements of the gear carrier relative to the banjo housing have been employed. In one design the center portion or ring of the banjo is horizontal and

the gear carrier bolts to the top of it. This is open to the objection that the axle housing will be weak at the center, owing to its small height, and will show considerable deflection. In another design, while the banjo ring is vertical and the gear carrier bolts against its forward face, the same as in Fig. 1, the intermediate shaft is located back of the axle shafts and somewhat higher. This arrangement is more compact than that of Fig. 1 and does not have so large an overhang. Overhang in front of the axle is not a serious matter, however, as it counteracts the torque-reaction on the axle housing during forward motion.

Center-Drop Axle—Fig. 5 is a vertical section through an axle used on Paris omnibuses. The particular object aimed

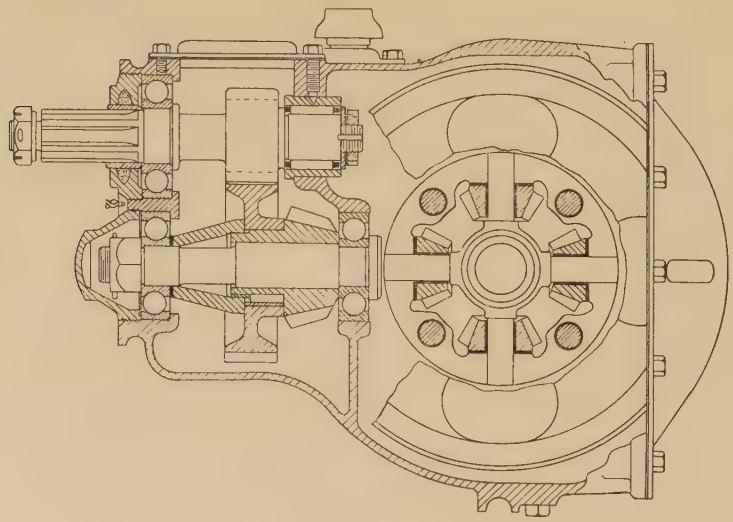


FIG. 4.—SPUR-BEVEL-GEAR AXLE.

at in this design is to have the center of the axle low, so it is not necessary to raise the floor over the axle in a bus of the safety or low-frame type. The axle housing consists of three castings, the central one forming a housing for the differential and bevel drive gears and also for the two pair of spur reduction gears which are located just inside the chassis springs. The end portions of the axle housings are tubes provided with large flanges at their inner ends by which they are bolted to the central casting. Both of the spur gears are straddle-mounted, and the axle shafts are relieved

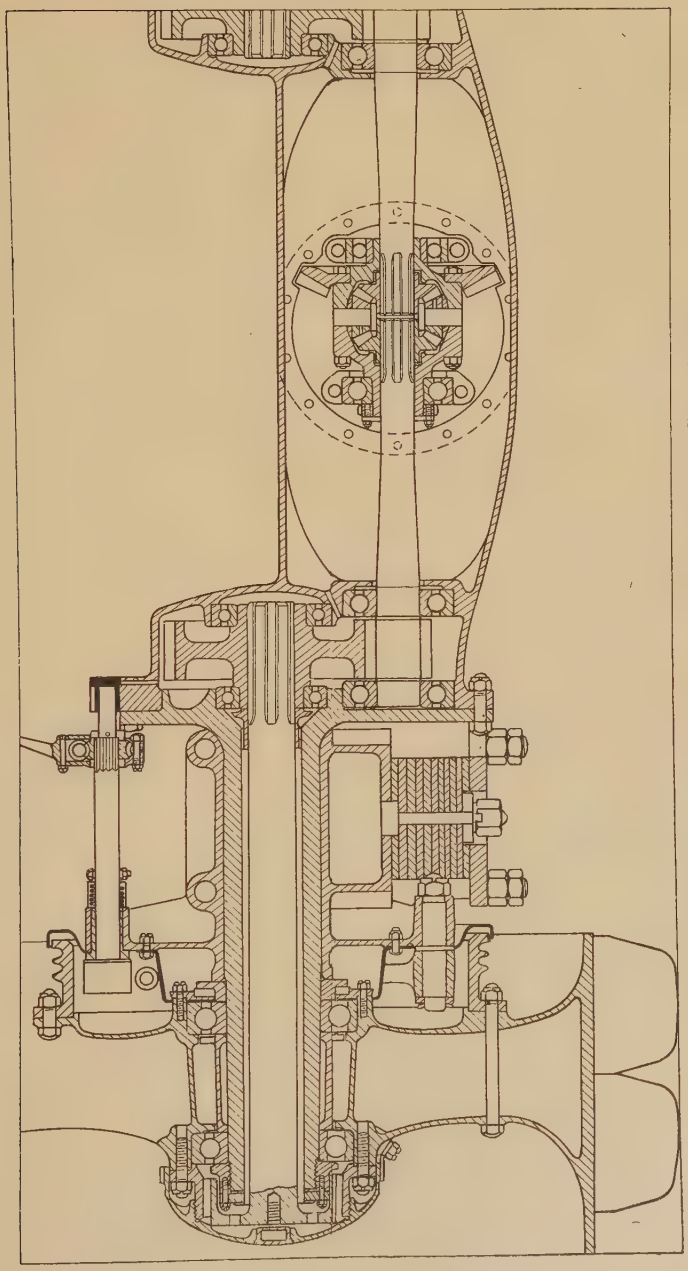


FIG. 5.—DROP-CENTER LIVE AXLE WITH BEVEL-SPUR-GEAR REDUCTION.

of all but torsional stresses, the axle being of the full-floating type. The spur pinions are formed integral with the differential shafts, and positive clutch members for driving the wheels are formed integral with the axle shafts. The other clutch member is a separate piece and is bolted to the cast steel wheel. It is threaded on the outside and has the hub cap screwed to it.

Internal Gears—At one time internal gears were largely used for the final drive of trucks and delivery wagons, but in recent years their popularity has decreased. This is probably accounted for chiefly by the fact that the speed of the lighter commercial vehicles has greatly increased, which has made it possible to use a single-reduction final drive for them, while in the field of heavier commercial vehicles the introduction of bevel-spur and similar drives has added further competition.

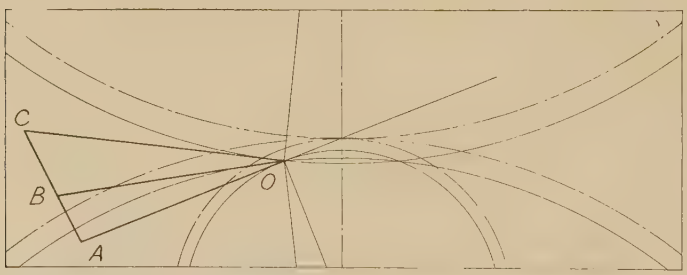


Fig. 6.—Relative Sliding Motions in Spur and Internal Gears.

It has always been claimed for internal gears that they are more efficient than any other type of gearing. This claim is based on the fact that in internal gears there is less sliding motion at the tooth contact surfaces than with spur gears, and far less than with helical and worm gears. This is shown graphically in Fig. 6.

Gear teeth have a combined rolling and sliding motion. When the teeth are in contact at the pitch point, since the contacting elements of both teeth are then moving in the same direction—along the common tangent to their pitch circles—there is only rolling motion and no sliding. But when contact is either ahead of or beyond the pitch point, the contacting elements move in different directions—each in the direction of a tangent to a circle through it and around the axis of its gear. The speeds of motion of the two elements are not equal. The speeds of the two pitch circles

are equal, but one of the contacting elements is below its pitch circle while the other is above it. The speed of each contacting element is equal to the product of the pitchline velocity into the ratio of its distance from the axis to the pitch radius. In Fig. 6 the various motions are represented for a pair of spur gears and for a spur pinion and an internal gear. Both pairs have the same pitch diameters, and the motions are shown for the same point along the line of action. O-A represents the motion of the contacting element of the pinion; O-B that of the contacting element of the internal gear and O-C that of the contacting element of the spur gear. The sliding motion between the teeth of the pinion and the internal gear is represented by A-B, while the sliding motion between the teeth of the pinion and of the spur gear is represented by A-C. There is therefore less sliding motion in the case of the internal gear; other things being equal, there will be less frictional loss in the case of this gear.

Location of Internal Gears—The internal gears are always connected to the road wheels, but there are two alternate arrangements. They may be located either on the inner side of the wheels, in which case they mesh with pinions at the outer ends of the differential shafts, or they may be on the outer side of the wheels, in which case they mesh with two or three intermediate gears which in turn mesh with a pinion at the end of each rear axle driving shaft, the pinion in this case being concentric with the wheel. Internal-gear rear axles with the internal gears on the inner side of the wheels may be again divided into two sub-classes according to whether they are enclosed by the carrying member or are outside of it. Since the driving shafts or axle shafts must necessarily be eccentric with respect to the wheel spindles, in order that they may be enclosed in the carrying member, the latter must be cranked. This construction lends itself particularly to use in connection with buses, as it permits of placing the floor of the bus body lower.

An internal-gear-type rear axle of the general type which has been most extensively used in truck work in this country, is shown in Fig. 7, the design being due to the International Harvester Co. The carrying member is of solid round section for the greater part of its length, but at the center it is dished to conform to the shape of the driving gear housing, which is telescoped into it and bolted against an annular finished surface on it. Just outside the spring seats the carrying member carries quite rigid flanged discs which serve both as brake supports and as bearing brackets for the bearings

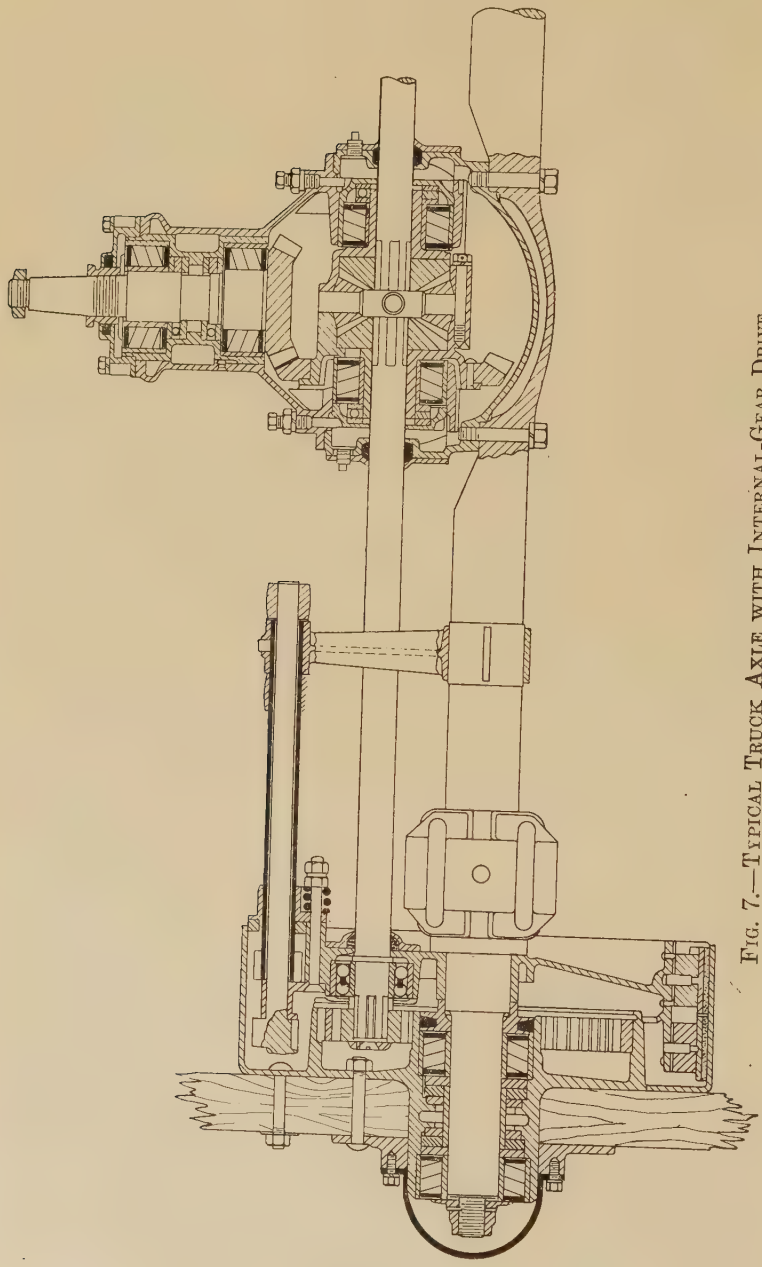


FIG. 7.—TYPICAL TRUCK AXLE WITH INTERNAL-GEAR DRIVE.

at the outer ends of the differential shafts. Between the driving gear housing and the bearings at their outer ends the shafts are exposed. The internal gear rings are fitted into drums cast integral with the wheel hubs.

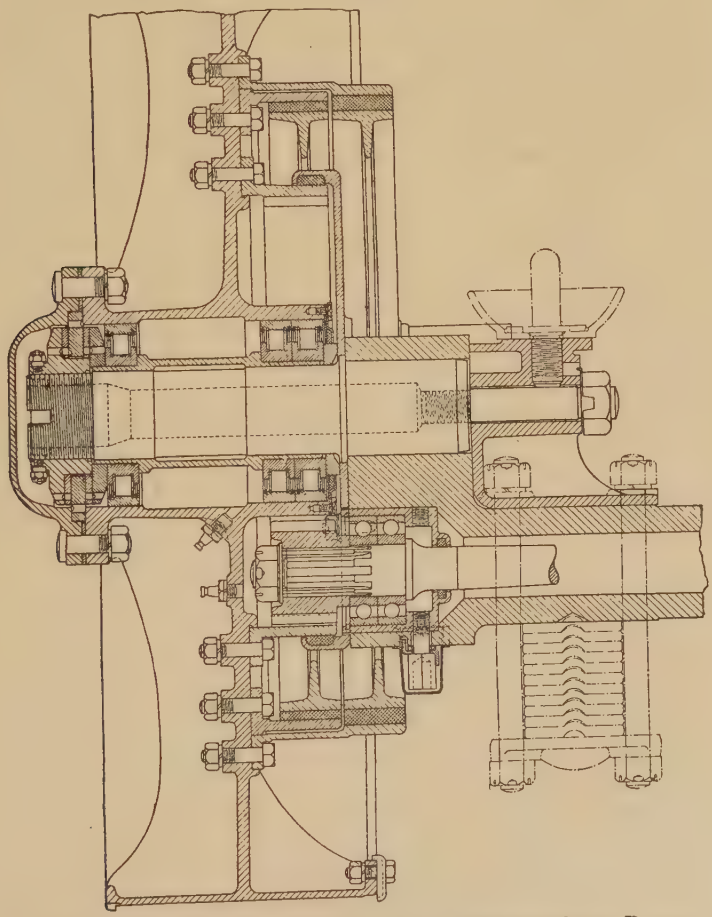


FIG. 8.—CRANKED REAR AXLE WITH INTERNAL-GEAR DRIVE.

The London General Omnibus Company in one design of bus has been using a cranked type of driving axle with internal gear drive, and the outer end of this axle is shown in section in Fig. 8. It will be seen that the axle spindle is a separate part and fitted into an eye in the end of the

cranked carrying member. The wheel bearings are cylindrical rollers, and end thrust in both directions is taken up on a double thrust bearing consisting of plain thrust washers at the outer end of the axle spindle. There are two annular ball bearings on the differential shaft at the pinion, mounted side by side inside an eccentric sleeve, which permits of close adjustment of the mesh of the gears. Gland packing is used outside the gear ring.

An example of the type of design in which the internal gear is located on the outside of the driving wheel is the Renault heavy truck axle illustrated in Fig. 9. In this axle

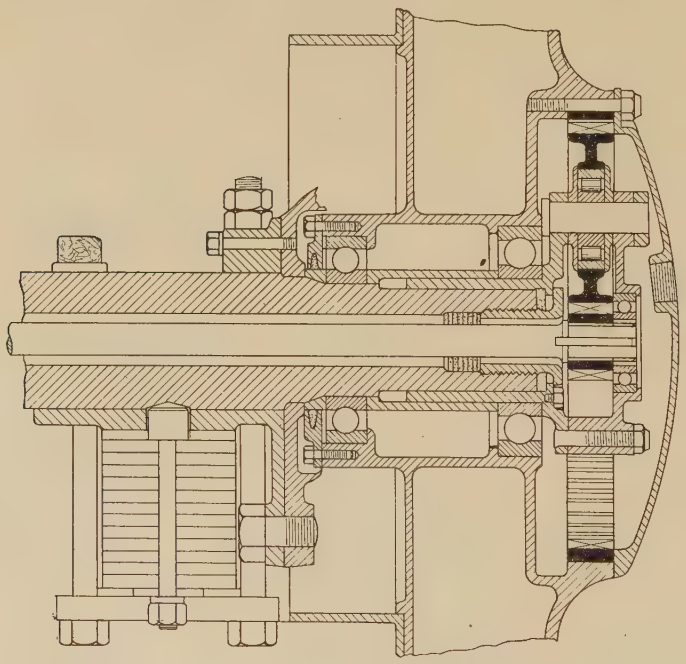


Fig. 9.—Rear Axle with Internal Gear on Outer Side of Road Wheel.

there is a bearing on the axle shaft outside the central pinion, this bearing being supported through a spider from the studs on which the planetary pinions revolve. In one design of axle of this general type the axle shaft was at first supported in a bearing mounted in the end of the axle tube, so that the piniion overhung its bearing. This, however, gave a great

deal of trouble and much better results were obtained by entirely eliminating the bearing and letting the pinion seek its position of equilibrium between the three intermediate gears in mesh with it.

An advantage of the type of axle in which the final speed reduction is effected at the wheels instead of at the center of the axle is that both the differential gear and the differential shafts can be considerably smaller and lighter.

Details of Internal Gear Drive—Where internal gears can be operated in a bath of oil and are rigidly mounted, they are unusually quiet, but, unfortunately, it is impossible to operate gears on the inner side of the road wheels in a bath of oil. Since the pinion shaft and the brake shaft have to extend through it, the cover for the brake and the gear must be stationary and there must be sliding joints between it on the one hand and the brake drum and gear ring on the other. These joints can be made so they are fairly dustproof, but they will not hold oil. In fact, since the internal gear is enclosed in the same housing with the brake, all lubricant must be kept away from it.

Since the pinion meshing with the internal gear (on the inside of the road wheel) overhangs its bearing, the load on the bearing is very heavy, and much difficulty has been experienced in the past with these outboard bearings. It is possible that in some cases this has been due to the actual bearing loads being considerably greater than the calculated loads owing to insufficiently rigid supports. Unusually heavy bearings are now used at this point (a single row ball bearing of the heavy series, for instance, with a diameter of the ball circle substantially equal to the pitch diameter of the pinion), and they are mounted as close to the pinions as possible.

Heavy-Duty Axle Ends—Figure 10 shows the outer end of a full-floating heavy-duty axle (Mack). The axle housing is of the dropforged banjo type and has integral spring seats. Inside the hub the wall thickness of the tube is considerably reduced and a combined brake carrier and cover plate is forced over the tube against a shoulder thereon. The end of the axle shaft is upset and splined, and power is transmitted from the shaft to the wheel by means of a driving clutch which is a free fit on the splined section of the shaft and is rigidly bolted to the wheel hub. The hub is mounted on two adjustable roller bearings on the outside of the axle tube. Any lubricant that may work out of the hub at the inner end (past the felt washer) is caught in an oil-return groove formed in the brake carrier and allowed to drop onto the road, thus preventing it from getting into the brake drum.

The driving clutch also forms a hub cap, the central hole in it being closed by a sheet steel disc. So-called Welch plugs are often used for this purpose.

The end of a semi-floating type of rear axle (White) is shown in Fig. 10. This has a cast housing which is notable for its heavy ribbing. There are two back-to-back taper roller bearings on the axle shaft at the wheel-end, whose inner races bear against a collar on the shaft and which take thrust on the wheel and on the shaft in both directions. One of the bearings is seated directly in an enlargement at the end of the

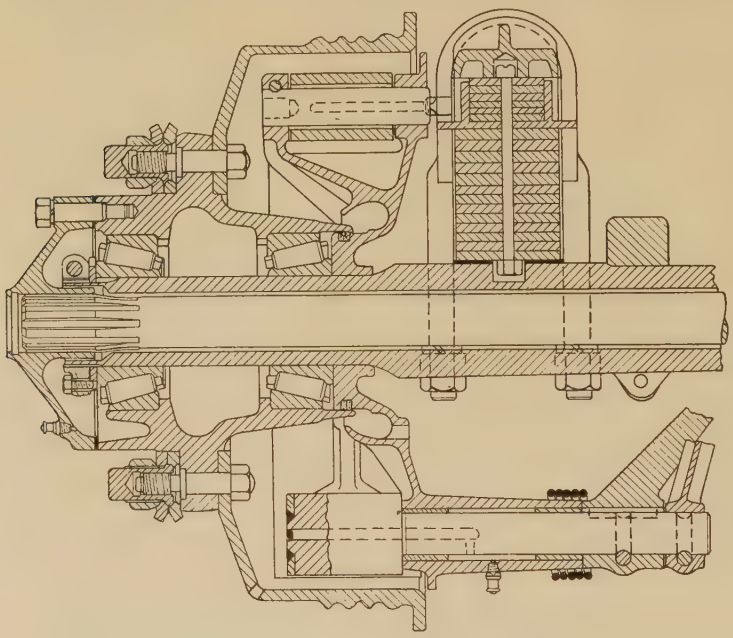


Fig. 10.—End of Full-Floating Heavy-Duty Axle.

axle housing, while the other is mounted in a cap which is piloted in the end of the housing.

The method of securing the wheel hub to the drive shaft is noteworthy. The inner end of the hub is tapered out and fits on a tapered section of the axle shaft, against which it is drawn by the nut on the end of the shaft. The outer end of the hub is internally splined and fits a corresponding splined section of the shaft. To ensure a tight fit of the splines, this section of the hub is tapered on the outside and

split longitudinally, and a tapered bushing is drawn over it by means of studs screwed into the hub flange and passing through holes in a flange on the bushing. The hub cap is screwed to this bushing.

Whatever the means employed for holding the wheel on the axle, they must be securely locked against coming loose, for in a high-speed vehicle the coming off of a wheel is a serious matter and must be thorougnly guarded against. In Fig. 10, it will be seen, there is a washer with a key or tooth on its inner circumference, which is passed over the tube first, the key extending into a keyway in the tube. The nut is then screwed on and locked, both by means of a clamp

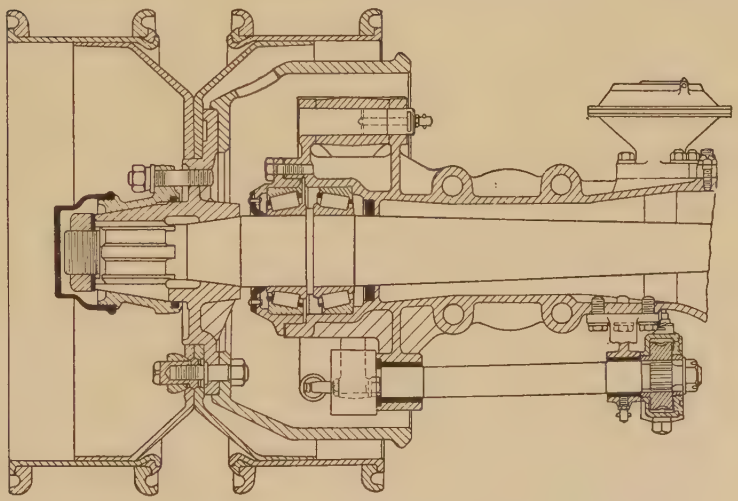


Fig. 11.—End of Semi-Floating Heavy-Duty Axle.

screw and a pin screw extending in to a slot on the washer, there being a series of slots extending all around the washer.

Calculation of Double-Reduction Axles—Assuming the gears to be made of No. 2320 steel or equivalent material, and considering that these axles are used only for the heavier classes of vehicle, the engines of which seldom run at more than 2000 r.p.m., we can allow a stress of 60,000 lbs. per sq. in. in the first reduction gears, based on the maximum torque of the engine and the low gear reduction in the transmission, and a stress of 70,000 lbs. in the second reduction gears. These figures are quite conservative and are often exceeded. The stress values are to be inserted in the formulae for spur

and bevel gears already given, and they can be used equally with spiral bevel gears, herringbone gears and internal gears, the calculation in every case being applied to the pinion. The formula for the calculation of spiral bevel gears given in the Chapter on Bevel Gear Driven Rear Axles is based on a considerably higher stress in the gear material, viz., about 90,000 lbs. per sq. in., but the load factor of a commercial vehicle engine is higher, for which reason it is well to use a somewhat lower stress. A similar figure (90,000 lbs. per sq. in.) would be used in commercial vehicles of the cheaper grade.

The capacity of gears is to some extent dependent upon the rigidity of their mountings. If the gears overhang their bearings they must not be made with too great a width of face, as, owing to shaft deflection, that portion of the face most remote from the bearing will be of little value. Straddle mounting is preferable for this reason. With bevel gears there is a tendency for the arms of the differential carrier to spread, and considerable trouble with differential bearings is due to this cause. This point should be kept in mind in designing the carrier.

If it is considered preferable to base the calculation of the rear axle gears on the engine torque at maximum horse power output, transmitted directly through the gear box, which approaches more closely to normal operating conditions, the stress in the first reduction and second reduction gears can be taken at 12,000 and 14,000 lbs. per sq. in. for the better grades of commercial vehicle, and 14,000 and 16,000 lbs. per sq. in. for the cheaper grades. Sometimes internal gears are not heat-treated after the teeth are cut. In that case the stress in the teeth should be limited to about 8000 lbs. per sq. in., based on maximum engine power and direct drive.

Calculation of Axle Tubes—In a general way the calculation of the stresses in axle tubes and of the necessary dimensions of such tubes is the same as in bevel-gear-driven axles, as explained in a previous chapter. In that case it was assumed that the axle had a central torque tube so that the reaction to driving and braking forces came at the center of the axle. Most truck and bus axles, however, employ the Hotchkiss drive (driving and braking reactions at center points of spring attachment), or have radius rods whose points of attachment to the axle are located close to the spring seats.

Let us assume that a drop-forged axle housing is to be designed for a 3½-ton truck. The ground reaction on each

rear wheel when the truck is fully loaded may be assumed to be 5000 lbs. (one-third of the total weight, loaded), and the distance from the center of the wheel to the center of the spring, 12 in. Assuming a maximum coefficient of road adherence of 0.6, the maximum traction or braking force on the wheel will be 3000 lbs. Hence, the maximum bending moment on the axle housing will be

$$\sqrt{(5000 \times 12)^2 + (3000 \times 12)^2} = 70{,}000 \text{ lb.-in.}$$

It is usual in the case of drop-forged axle housings, since they must be bored out from the solid, to make the inside diameter only slightly larger than the diameter of the upset ends of the axle shaft. These upset ends in the present case probably would have a diameter of 2 in., and we can make the bore of the housing $2\frac{1}{16}$ in. The question then arises, What must be the outside diameter of the axle between springs? Since the housing is of drop-forged, heat-treated alloy steel (nickel or chrome nickel) a working stress of 20,000 lbs. per sq. in. can be allowed. We then have

$$70{,}000 = \frac{20{,}000\, I}{c} \text{ and } \frac{I}{c} = 3.5,$$

where I is the moment of inertia of the section of the axle tube and c the distance from its axis to its outermost fiber (that is, one-half the outside diameter. Let x be the outside diameter required. Then, since the moment of inertia of a hollow circular section of outside diameter d and inside diameter d_1 is

$$\frac{\pi\,(d^4 - d_1{}^4)}{64},$$

we have

$$\frac{\dfrac{3.1416\,(x^4 - 2.06^4)}{64}}{\dfrac{x}{2}} = 3.5,$$

which gives

$$0.0491\, x^4 - 0.28 = 1.75\, x.$$

Solving by trial and error we get,

$$x = 3.35 \text{ (say, } 3\tfrac{3}{8} \text{ in.)}$$

Drop forged rear axle housings are made of such steels as Nos. 2335 and 3140, which, when heat-treated to show a hardness of about 300 Brinell, have an elastic limit of about

130,000 and an ultimate tensile strength of about 150,000 lbs. per sq. in. Hence, the factor of safety is about 7.5. A similar factor of safety should be used with other materials, such as cast steel and malleable cast iron. This means that with cast steel the working stress can be made 8000 to 10,000 lbs. per sq. in., according to quality, and with malleable iron, from 6000 to 8000 lbs. per sq. in. In general it is advisable to use a somewhat lower working stress in axles for solid-tired vehicles.

Cast housings are made of square section, as a rule, primarily because this facilitates the use of separate spring seats, which are convenient when the axle is made by a specialist and spring centers have to be varied for different customers. A hollow square section, however, also weighs less than a hollow round section of the same outside diameter and the same section modulus; and, finally, the pattern for a square-section axle housing is somewhat easier to make than that for a round one.

CHAPTER X

Front-Drive, Four-Wheel-Drive and Six-Wheeled Vehicles

For certain special purposes other types of vehicle than the conventional one with two front driving and two rear steering wheels are more desirable. For instance, for racing cars for use on circular tracks, front-wheel-drive has met with considerable favor. As compared with a rear-driven vehicle the front-driven track racer has the advantage that it is less inclined to skid at turns. Moreover, since the propelling effort is always in the direction in which the vehicle is headed, the application of power on turns is probably more efficient. Finally, since there is no propeller shaft to the rear axle, the driver's seat and other parts between the engine and rear axle can be placed lower, and the stability of the vehicle thus increased.

Front-wheel drive has been introduced also for stock passenger cars. The chief consideration in this case is that it permits of placing the floor of the car lower, and thus lowering the center of gravity and increasing the stability. The same advantage holds in bus work, and is of particular importance in connection with double-deck buses, in which the center of gravity is naturally quite high when they are loaded. There is one disadvantage connected with front-drive that must not be overlooked, and that is that the reaction to the driving torque virtually transfers some of the weight from the front to the rear wheels, thereby reducing the ground adherence of the front wheels and the maximum torque which can be transmitted through them.

When a car is being driven forward the pressure of its front wheels on the ground is not as great as when it is standing still, and the ground adherence of the front wheels therefore also is less in the former case. The ground pressure and ground adherence of the rear wheels increase in the same measure as the corresponding values for the front wheels decrease; therefore, the maximum torque which can be transmitted through the front wheels is less than that which can

be transmitted through rear wheels carrying the same static load. Moreover, in the case of a rear drive it is easy enough to increase the load on the rear wheels by extending the frame behind the axle, and although a similar expedient may be resorted to in the case of the front drive, the result is not a happy one from the esthetic point of view.

When the front wheels rotate in one direction with a certain torque, the reaction of the drive tends to make the car frame turn around the front wheel spindles in the opposite direction with the same torque. A little reflection will make it clear that this torque reaction tends to press the rear wheels harder against the ground, and in the same measure as the pressure of the rear wheels against the ground is increased, that of the front wheels is decreased. Let P be the propelling force of the front wheels and r the front wheel radius; then, if the length of wheelbase is w, the amount of weight which is virtually transferred from the front to the rear wheels due to the torque reaction is Pr/w. The maximum value of P, the propelling force of the front wheels, depends upon the load L on these wheels and may be assumed to be 0.6 L. As in an average passenger car the ratio r/w is of the order of $\frac{1}{8}$, substituting for P and r/w in the above expression for weight transfer we find it to have a limiting value of about 0.075 L or 7.5 per cent of the weight on the front wheels. The situation is somewhat aggravated, however, by the fact that the maximum propelling effort is required on steep upgrades, on which there is a further transfer of weight from the front to the rear wheels due to the inclination of the vehicle.

To allow for this unfavorable shifting in weight due to driving torque it is advisable to so arrange the general layout that the center of gravity will be farther forward than in the conventional rear-driven car.

Arrangement of Front Drives—The general plan in connection with front-wheel-driven passenger cars is to build the engine, clutch, transmission and final drive with differential gear in a single unit which is mounted on the frame at the forward end but in the reverse direction, the driving end extending forwardly. The axis of the differential is located in the vertical plane through the steering pivot axes, and from the differential, which is thus supported on the chassis springs, connection is made to the front wheels by means of shafts incorporating universal joints—one joint near the differential housing and another one placed symmetrically with respect to the steering knuckle pin axis. The latter joint must meet unusual requirements, for it must take care of

changes in the angularity of the shafts connected, resulting not only from spring action but also from the steering motion. The change in angularity due to spring action can be readily taken care of, for it affects the universal joints at the differential and at the wheel end alike, and it is only necessary to have the yokes (or their equivalents) at opposite ends of the intermediate shaft in the same plane in order to prevent the occurrence of abnormal inertia stresses. Deflection of the steering wheels, on the contrary, affects only the universal joint at the knuckle pin. If a single joint of the conventional type were used there, since the steering deflection may be as great as 40°, the power would be transmitted to the wheels with great irregularity when making sharp turns, and the consequent strains on the transmission mechanism and the tires would be severe. It is, therefore, customary to use either a single joint of the type transmitting motion angularly without fluctuations, or else two ordinary joints separated by a very short intermediate shaft and disposed symmetrically with respect to the knuckle pin axis. The single universal joint must be so located that its center lies in the axis of the steering knuckle pin.

An incidental advantage of front-wheel-drive is that the front wheels may be utilized for braking without mounting brake drums on them, drums so located being undesirable because they increase unsprung weight and promote forms of front-end periodic vibration known as wheel shimmy and wheel wobble. Front-wheel braking can be provided for by placing a brake drum on the propeller shaft or on each of the differential shafts where they emerge from the differential housing. This construction, of course, also eliminates the somewhat complicated operating linkage to brakes turning with the steering motion of the wheels.

Front-wheel drive has been used for certain special types of commercial and municipal vehicles. Many horse-drawn hose-and-ladder wagons were converted to the more economical motor-drive by replacing their original fore-carriage by a motor fore-carriage, and front-driven trucks are in use where a very low floor height is desirable, as in transporting race horses and other heavy animals. These trucks have deeply cranked rear axles which permit of placing the floor within a foot or so of the ground.

Four-Wheel Drives—The four-wheel drive possesses particular advantages for military trucks and tractors which frequently must operate away from beaten roads. During the War the Ordnance Department of the U. S. Army bought large numbers of four-wheel driven trucks, and it also devel-

oped a design of its own. The drive is also adapted for trucks for use in mining work and for haulage in undeveloped countries. Since the advantage of the four-wheel drive is primarily due to the fact that all of its weight is available for traction purposes, it lends itself particularly for use on vehicles designed to haul one or more trailers behind them. Aside from its greater limiting traction, the four-wheel-driven truck has the advantage that the total load can be divided substantially equally between its four wheels, which obviates abnormal loads on any wheel and makes unnecessary an excessive overhang over either axle. For general haulage on well-paved roads the use of four-wheel-drive vehicles does not seem to be warranted, for their cost of construction is necessarily higher, and their advantages do not come into play there.

The problem of a four-wheel drive consists chiefly in combining the functions of steering and driving in a single axle. As the steering wheels swing around a substantially vertical axis for steering, a universal connection between the wheels and the propelling mechanism on the frame is necessary, unless the engine and drive are both supported on the axle, or on that part of it which swings in steering. The problem is about the same as that involved in the design of front wheel drives.

One method of transmitting power to pivoted front wheels makes use of a train of bevel gears, of which a pair of intermediate gears is mounted concentric with the steering knuckle pin. This arrangement has some important advantages, but it is rather complicated. In one French design of a four-wheel driven truck employing this construction no less than sixteen bevel wheels are used in the drive between the gear box and the front wheels. One of the good points of this type of steering and driving axle is that, since the bevel gears secured to the road wheels swing around the knuckle pivots, no irregularity in the transmission of motion is introduced. That is, whether the road wheels are in the straight-ahead position or not, the transmission of motion from the tail shaft of the gear box to the road wheels is always effected at a constant ratio, there being no periodic fluctuation as with ordinary universal joints.

In another construction universal joints are employed to transmit the motion from shafts concentric with or parallel to the axle to a pair of short shafts carried by the steering knuckles on opposite ends of the axle. These four shafts, viz., the two long differential shafts and the two short shafts connected to them, may be either concentric with the axle, in which case the short shafts connect to the wheel hubs

through driving dogs, or they may form a countershaft, in which case the short shafts carry spur pinions which mesh with spur or internal gear crowns on the road wheels. In either case the universal joint center must lie in the axis of the steering knuckle pin.

Drop Gear—In four-wheel drive trucks the engine is generally carried in a comparatively high position, so that the propeller shaft to the front axle can be carried under

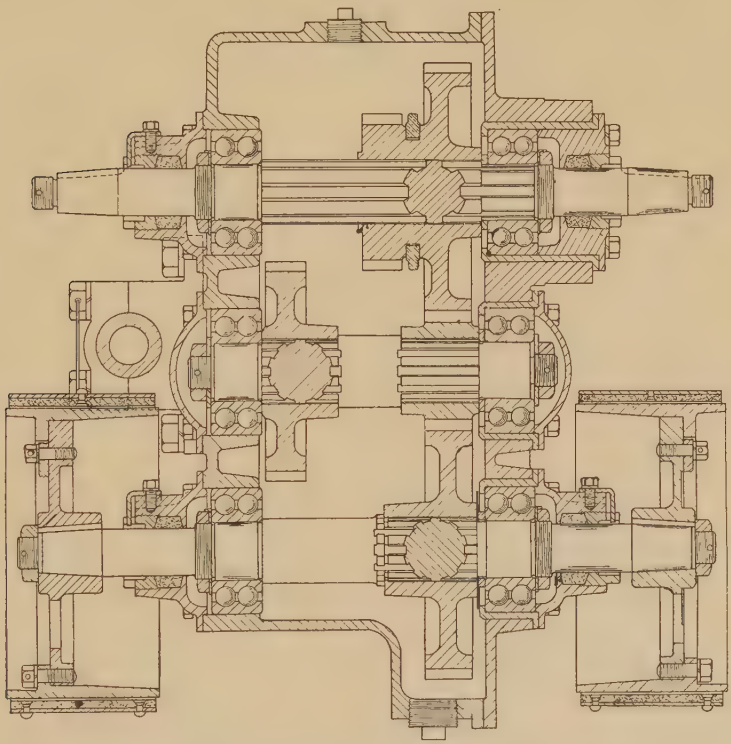


FIG. 1.—DROP GEAR OF FOUR-WHEEL-DRIVE TRUCK.

it, or under its supporting arms, without interference. This calls for the use of a "drop gear" by which the power is transmitted to a lower level than that of the transmission drive shaft. Such a drop gear (Coleman) is illustrated in Fig. 1. The power comes in from the transmission at one end of the upper shaft and is transmitted through spur gears to an intermediate shaft and thence to the lower shaft, to the

ends of which the two propeller shafts connect through universal joints. This particular drop gear also forms an auxiliary transmission. With the gears in the position shown, the power is transmitted from the top to the bottom shaft without change in speed. By sliding the pair of pinions on the top, splined shaft to the left until the smaller pinion is in mesh with its gear, a speed reduction of about 3:2 is obtained.

It will be seen that in this truck the power is transmitted to the front and rear axles without passing through a differential gear. There are some arguments for the use of a differential gear between the two axles, but these also apply in the case of a six-wheeled vehicle with two driven axles, and the problem will be discussed in connection with that subject.

Figure 2 shows the drive to the front wheels of a British War Office four-wheel-drive truck built by Thornycroft. Both axles have worm drive. At each end of the front axle housing a vertical, semi-cylindrical chamber is formed, which is closed by a corresponding part formed on the steering knuckle, the sliding joint between the two being rendered fairly grease-tight by means of felt packings in grooves formed in the part on the knuckle. At the end of the axle drive shaft there is a bevel pinion which meshes with an idler gear mounted on ball bearings on the knuckle pin, this idler in turn meshing with a bevel gear concentric with and bolted to the wheel hub.

A combined driving and steering axle, in which a universal joint is placed inside the forked axle end, is manufactured by the Four Wheel Drive Auto Co., Clintonville, Wis., and a part sectional view of this axle is shown in Fig. 3. On the truck of this concern the change speed gear is located at the middle of the frame, and from the transmission drive shaft of the gearset the power is transmitted by a silent chain to a fore-and-aft shaft with differential gear and universal joints. From this fore-and-aft shaft the power is transmitted to the two driving axles by bevel pinion and gear, a sufficiently large gear reduction being obtainable because the speed is already reduced somewhat by the silent chain connection between the gear box and the longitudinal shaft. Each axle contains one differential gear, so that there are three in all on the truck.

Details of the construction of the steering and power transmission joint are shown in the illustration. The steering knuckle pivot is developed into a spherical joint which contains trunnions, fully enclosing the universal joint. This joint is of compact design, and as it has to transmit only one-fourth of the power of the engine it is amply large for its purpose.

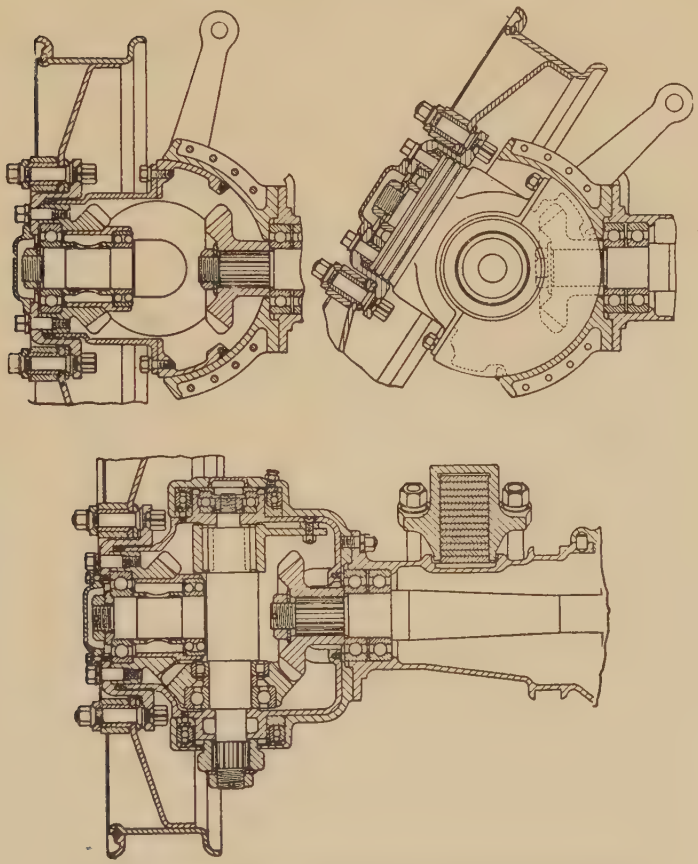


FIG. 2.—FRONT-WHEEL DRIVE THROUGH BEVEL GEARS.

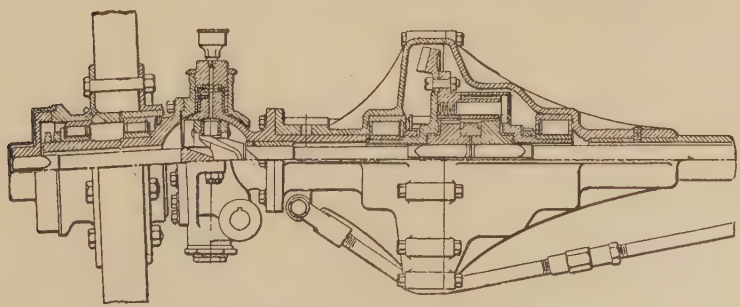


FIG. 3.—FOUR-WHEEL DRIVE AUTO CO. FRONT AXLE.

Six-Wheelers—Six-wheeled vehicles were introduced about 1918 in connection with a movement to promote the use of pneumatic tires on heavy vehicles. They permit of transporting greater loads on a single vehicle than is possible with a four-wheeler. It was found in the course of experimental work with giant pneumatics, that 10 in. is about the largest size that can be satisfactorily used in road transportation. Tires of 12 in. width were built, but they are too large to be handled on the road by one man, in case of a puncture or blow-out. Thus the total weight of a six-wheeled vehicle with load, if pneumatic tires are to be used, must not exceed the carrying capacity of six 10-in. tires. By using twin tires on the rear wheels the same total load could be carried on a four-wheeler, but tests by road departments brought out the fact that the maximum ground pressure is much less if a given load is divided between two pairs of driving wheels a certain distance apart in the direction of the axis of the vehicle, than if it is carried on a single pair of wheels (see

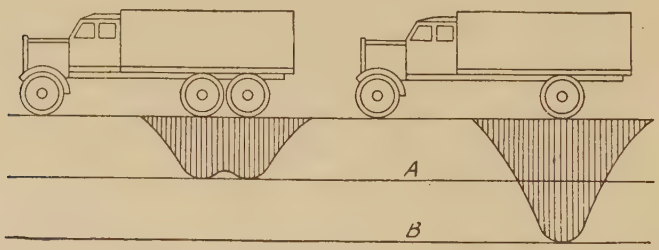


FIG. 4.—STRESS DISTRIBUTION IN ROAD BED UNDER REAR WHEELS OF FOUR-WHEELED AND SIX-WHEELED VEHICLES

Fig. 4). Various state legislatures therefore raised the load limit for six-wheelers above that for four-wheelers, and this stimulated the demand for six-wheelers.

In England the use of six-wheeled construction was encouraged by the decision of the War Office to pay a subsidy to purchasers of six-wheeled trucks meeting certain specifications rendering them suitable for military purposes. A design ensuring great flexibility of the chassis, enabling it to negotiate rough roads without injury, was developed and patented by the War Office technical staff, and its use was offered free to manufacturers wishing to manufacture "subsidy lorries."

For motor bus work the six-wheeler has the additional advantage of easier riding qualities. Each driving wheel of

a six-wheeler naturally weighs much less than one driving wheel of a four-wheeler of equal total weight; hence, when one wheel meets an obstruction and is thrown upward, the reaction on the chassis and body is less in the case of a six-wheeler. Moreover, the system of levers represented by the springs and axles by which the upward movement of the wheel is transmitted to the chassis frame has a greater reduction ratio in the case of the six-wheeler. Of course, the chassis of the six-wheeler receives a larger number of shocks in a given time, but this does not nullify the advantage gained through the reduction of the magnitude of the shocks, for if we could continue to increase the frequency and decrease the magnitude of the shocks indefinitely, we would reach an ideal condition. It is also claimed for six-wheelers that they are less given to skidding than four-wheelers, and this claim seems logical, for there is naturally less chance of four wheels losing their grip on the road at the same time.

Effect of Unequal-Sized Wheels—In a six-wheeled vehicle the two rear axles generally are driven axles. They can be either positively geared together, or they may be driven through a differential. If the mean diameters of the two pairs of driving wheels were exactly equal, there would be no need for a differential; both wheels must progress along the road at the same speed, and if they are both of the same diameter, equal road speeds correspond to equal speeds of rotation. But in practice the wheel diameters never are exactly alike, differences being due chiefly to different degrees of tread wear and different degrees of inflation. What counts, of course, is the effective diameter, as determined by the distance from the wheel axis to the ground, which is less than the normal diameter by reason of the compression of the tire under load.

If the effective diameter of one pair of driving wheels is x per cent less than that of the other, the smaller wheels must slip over the ground in the direction in which the vehicle is progressing a distance equal to x per cent of that moved by the vehicle. That the smaller wheels slip, rather than the larger ones, is due to the fact that the driving gears form a sort of equalizing gear which balances the moments produced by the ground reactions on the two pairs of wheels, acting on lever arms equal to the mean radii of the two pairs of wheels, respectively. If we assume the weights on the wheels and their ground adherences to be equal, respectivly, the limiting moments of the larger wheels will be greater, owing to their greater lever arms, and slipping therefore will occur first at the smaller wheels. The larger wheels then

will produce not only the entire propelling force required to overcome the rolling resistance of the vehicle, but in addition the propelling force necessary to overcome the resistance to forward slippage of the smaller wheels.

Now let us suppose that the ground adherence of the larger wheels is not sufficient to produce the propelling effort to overcome both resistances. Then these wheels will slip in the direction opposite to that of motion, and while so doing will continue to produce the maximum possible propelling effort in the direction of forward vehicular motion. The remainder of the propelling effort required will be produced by the smaller wheels, which under these conditions will not slip. If the resistance to motion is further increased, a point is finally reached where both sets of wheels will slip in the direction opposite to that in which the motive power tends to propel the vehicle.

In the above discussion it has been assumed that all four of the driving wheels press down upon the road with the same pressures and that the ground adherences of all are the same. This condition may be approached on smooth roads at low speeds. Under normal conditions, however, when driving at some speed over moderately rough roads, the different wheels successively are thrown upward by road obstructions and partly or completely lose their ground adherence. Therefore, whereas when driving slowly over smooth roads with a large coefficient of adherence all of the slippage will occur at the smaller wheels, and the drive will be entirely by the larger ones, on rough roads the required slippage generally will occur at the wheel having momentarily the least ground adherence. The ground adherence of one of the larger wheels must drop below that of the smaller wheels having th lowest adherence by a percentage equal to percentage difference in effective diameters before the large wheels begin to slip. When there is slippage of the larger wheels these wheels will produce the maximum forward propelling effort they are capable of, and the remainder of the propelling effort required will be produced by the smaller wheels.

Loss of Power Due to Slippage—Slippage of one pair of wheels results in loss of power and, of course, in wear of the tires. We found that the percentage of slip is equal to the percentage of difference in wheel diameters. In order to get an idea of the loss in power represented by this slippage, let us assume that in a six-wheeled vehicle 30 per cent of the total load is carried by the front steering wheels, and 35 per cent by each pair of the rear driving wheels; also, that the

rolling resistance is 20 lbs. per thousand (resistance coefficient 0.02), and the friction coefficient between the tire and ground, 0.6. We will designate the total weight of the vehicle by W. Then the resistance to rolling motion is $0.02W$. The resistance to the sliding motion of the smaller driving wheel is

$$0.35W \times 0.6 = 0.21W.$$

If the smaller wheels have an effective diameter 1 per cent less than that of the larger wheels, the sliding motion is only one-hundredth the rolling motion. The propelling force that overcomes the sliding motion originates at the same point as that which overcomes the rolling motion. At this point, owing to the much higher speed, a force only one-hundredth as great as the resistance to sliding, is required to overcome the resistance to sliding. The force necessary at the circumference of the larger wheels to overcome the resistance to sliding at the circumference of the smaller wheels therefore is

$$\frac{0.21W}{100} = 0.0021W,$$

or about 10 per cent of the total propelling effort required. This applies where roads are comparatively smooth. Where roads are rough and speeds comparatively high, one or another of the four driving wheels will be nearly off the road practically all the time; it is this wheel that will slip and the loss due to slippage will be proportional to its ground adherence and therefore will be small. Only one wheel of a pair needs to slip, owing to the differential between each pair of wheels.

Need for a Third Differential—For the above reasons the need for a differential between driving axles is greatest in the case of low-speed vehicles operating on comparatively smooth roads. Certain six-wheeled buses in service in London are fitted with a differential between axles, while most of the trucks and buses of the six-wheeled type that have been placed on the market are without it, the two driving axles being positively geared together. In the operation of the latter type of vehicle it is desirable to select tires so that the mean diameters of the two pairs on the rear wheels are as nearly alike as possible. Records of tire mileages with such vehicles indicate that there is no excessive sliding of their tires under load.

Types of Six-Wheelers—Figure 5 illustrates what was probably the first practical six-wheeler with four driving wheels. It was developed by the Goodyear Tire & Rubber Co.

of Akron, O., beginning in 1918, and was later manufactured by the Six-Wheel Corporation of Philadelphia. The drive to both axles is by worms and wheels, the two worm shafts being connected together by means of an intermediate shaft with universal joints. Torque reaction is taken up on two telescoping tubes between the two axles, each tube being connected to one of the axle housings by a vertical pivot joint. The drive is transmitted through half-elliptic springs which extend between the axle tubes and carry the frame at their middle.

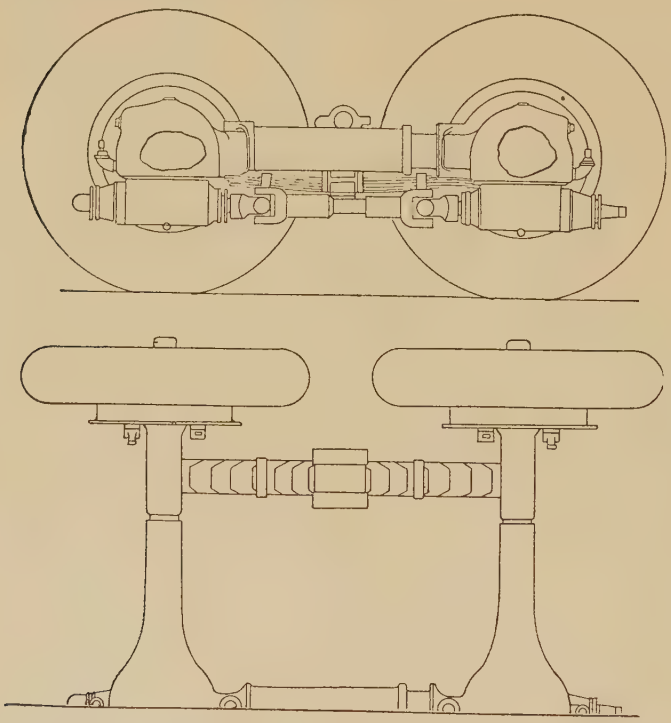


Fig. 5.—Rear Construction of Safeway Six-Wheeled Bus.

It was pointed out in connection with front drives that when driving torque is applied to either the front or rear axle, weight is virtually transferred from the front to the rear axle. This assumes that the torque reaction is taken on an arm of which the outer end is supported on the frame. In a construction like that shown in Fig. 5, where the two

axle housings are connected together by a member taking the reaction to the driving torques, weight is virtually transferred from the front driving axle to the rear one whenever driving torque is applied. Since the two axles are placed as close together as possible (in order to reduce to a minimum the unavoidable side slip of the rear wheels in turning cor-

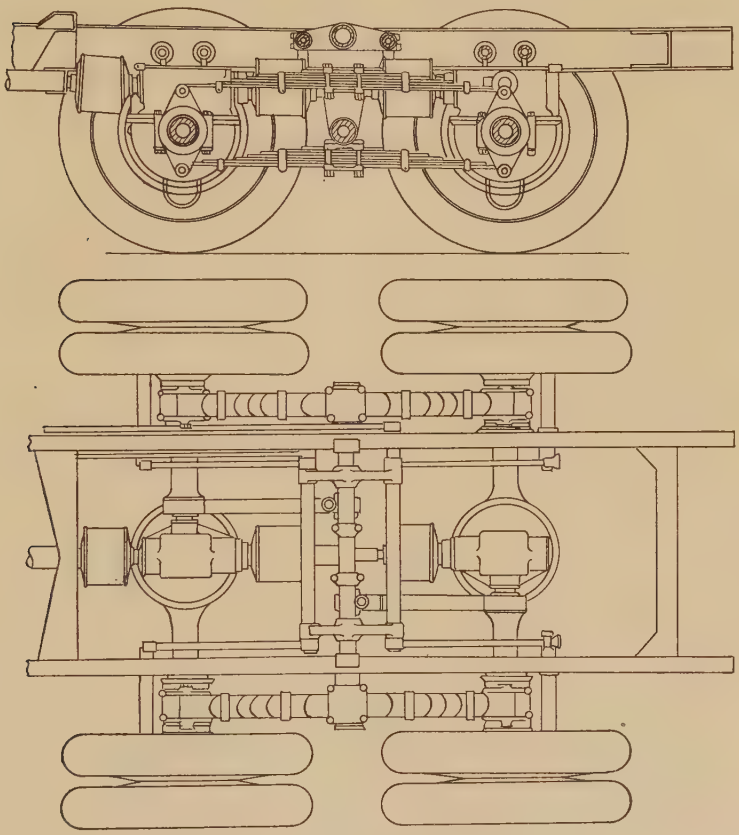


Fig. 6.—Rear Construction of British War Office Six-Wheeler.

ners), this weight transfer is comparatively large. It is advisable to allow for it in the design of the vehicle, by so arranging the rear springing that when no driving force is applied the load on the forward pair of wheels is about 10 per cent greater than that on the rear pair.

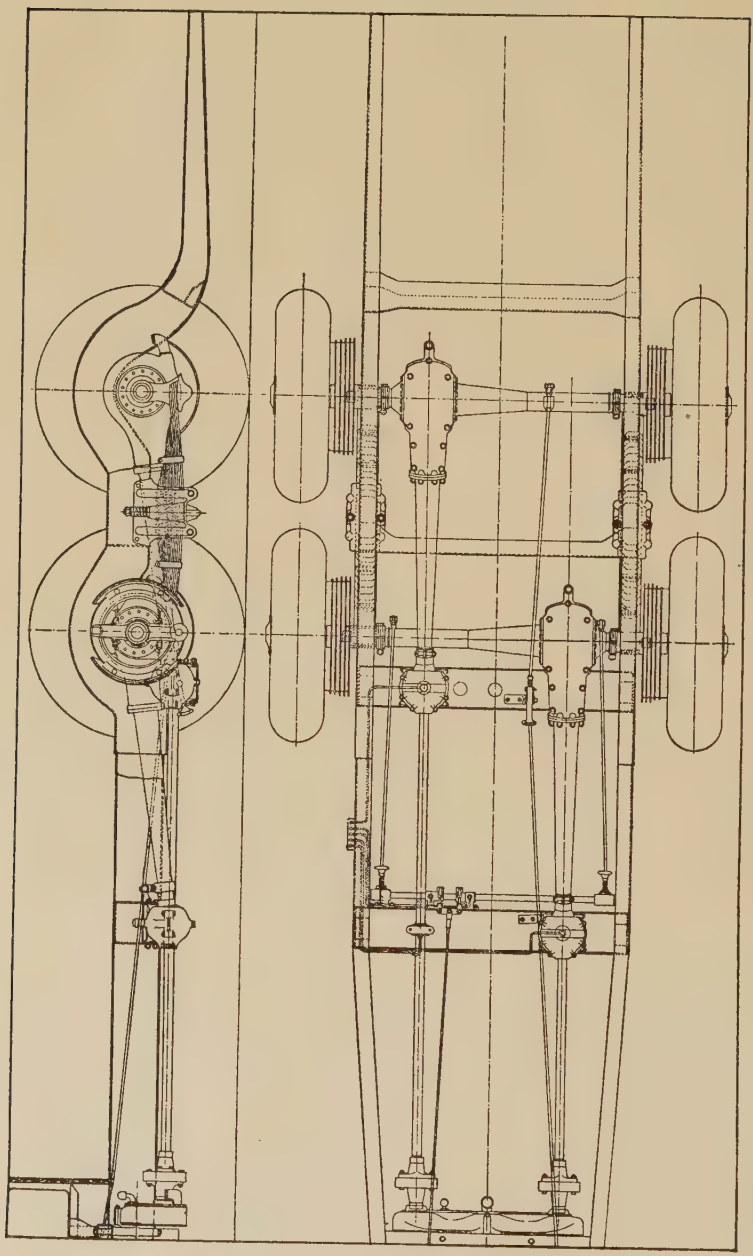


FIG. 7.—REAR CONSTRUCTION OF BUESSING SIX-WHEELER.

A design in which the torque reaction is taken up by telescoping tubes between axle housings has been used also by Renault in France. The Renault six-wheeler has bevel-gear drive, the pinion for the forward axle being located back of the axle, and that for the rearmost axle, in front of it. The two pinion shafts are connected by a short shaft with one sliding and two universal joints, the power being transmitted to the forward pinion shaft through spur gears. There are two sets of spur gears, these forming an auxiliary two-speed transmission. The telescoping tubes form the housing for the auxiliary transmission and the shaft connecting the bevel pinion shafts.

British War Office Design—Reference has been made already to the six-wheel-drive construction of the British War Office, and this is illustrated in Fig 6. One feature is that the torque on each axle is taken on a short torque rod extending to a frame cross member located between the two axles. With such an arrangement of the torque members the load on each rear axle is added to equally if both exert an equal driving torque, as they will if there is a differential between them; otherwise, if the mean diameters of the two pairs of wheels are not alike, the load on one driving axle will be added to by the torque while that on the other will be reduced by the negative torque due to the friction of slippage. This is an advantageous feature, since it is desirable to have additional load on the wheels doing the driving, whereas the load on the slipping wheels should be kept down, because that reduces the losses.

There are two pairs of half-elliptic springs in this six-wheeler, which virtually form parallel links. These springs are pivotally mounted on frame brackets, and the whole structure is very flexible, enabling the vehicle to accommodate itself to very uneven road surfaces. The difference in the levels of the two axles may attain 9 in., and the angles between the two worm shafts and the intermediate shaft, 26°, hence universal joints capable of operating at large angles must be used.

Buessing Design—Many different six-wheel constructions have been worked out, but this type of vehicle may still be regarded as in its developmental stage. In a German design by Buessing (Fig. 7) there is a spur pinion at the rear of the transmission which meshes with two spur gears on opposite sides of it. From the shafts of these spur gears power is transmitted independently to the two driving axles by propeller shafts and bevel gears, the differential housings of the two axles being located off-center and the propeller shafts suf-

ficiently inclined so there will be no interference between the forward driving axle and the propeller shaft to the rearmost one. The drives are of the enclosed propeller-shaft type and both driving thrust and torque reaction are taken on the propeller-shaft tube. This relieves the springs of all auxiliary functions and their ends slide on wear plates underneath the axle housings.

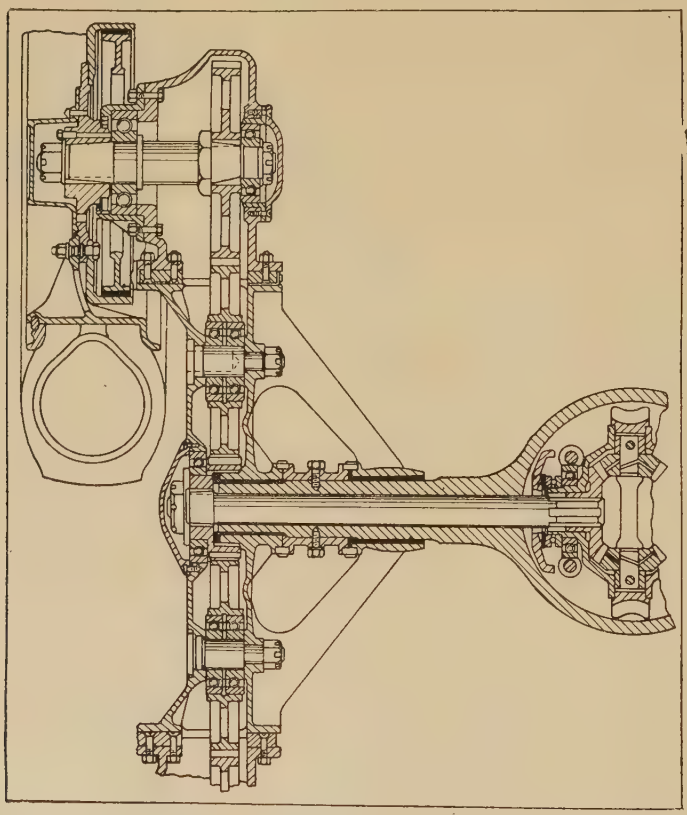


Fig. 8.—Scammel Single Rear Axle with Four Driving Wheels.

Single Driving Axle with Four Wheels—Scammell Lorries, Ltd., in England have produced a heavy truck having a single rear driving axle but four rear driving wheels, as shown in Fig. 8. On each end of the axle tube is supported a housing which contains a gear train connecting two wheel

shafts supported in bearings in it, with the axle shaft. The gear housing swivels on the axle tube and is provided with a diagonal brace giving it adequate strength to resist moments due to one of the wheels receiving a lateral shock. One advantage claimed for this six-wheeler is that the wheels are free to adapt themselves to exceedingly uneven road surfaces. Moreover, the wheels on each side always remain in the same plane, in contrast to what happens when a vehicle with two driving axles travels over rough ground. This is of some importance in that it improves the working conditions for a non-skid band or chain track over the two wheels.

CHAPTER XI

Brakes

The brakes of a motor car serve three distinct purposes. They are used to bring the car to a stop rapidly, to prevent it from attaining too high a speed when descending steep down-grades, and to hold it in position on grades. Motor car brakes may be divided into two classes, viz., drum brakes and disc brakes. Drum brakes may again be divided into internal and external (or expanding and contracting) brakes. Every brake consists of two members or sets of members, one of which is secured either directly to a road wheel or to a shaft in permanent driving relation with a road wheel (so that it rotates whenever the car is in motion), while the other is supported on the car frame or axle in such a way that it cannot rotate. Means are provided for bringing the two members into frictional engagement ("applying the brake"). Supposing the car to be in motion, when the brake is applied, the friction between its two members must be overcome. This results in the absorption of energy, and that energy is withdrawn from the store of kinetic energy of the vehicle, whose speed is thereby reduced.

Legal Requirements—Practically all motor vehicle laws call for two independent brakes, and sometimes it is stipulated that at least one of these brakes must act directly on the road wheels. The idea behind this legal requirement is that no mechanism is proof against failure, and if there were only a single brake and that should fail, the operator would be helpless. The second brake, therefore, was intended for emergencies only. The terms "service brake" and "emergency brake" have come into wide use in this country, while in Europe the terms "foot brake" and "hand brake" are more common.

Service brakes are generally designed for most convenient operation, by means of a pedal, and so as to be capable of withstanding continuous use over a long period, whereas the emergency brake, which in the majority of cars is operated

by means of a hand lever, often is so proportioned that, although it is nearly as powerful as the service brake in bringing the car to a stop, it cannot endure continuous service for long. Recently service brakes have been so much improved that their complete failure is only a very remote possibility, and the term "parking brake" has been applied to the second brake, to indicate that it is intended principally for holding the car in position. The hand-operated emergency or parking brake can be set in the "on" position by means of a latch, while the pedal-operated service brake can not.

Types of Brake—The drum-type of brake may be either a band brake or a shoe brake. Both band brakes and shoe brakes may be either internal or external, though, as a rule, band brakes are external and shoe brakes internal. The sta-

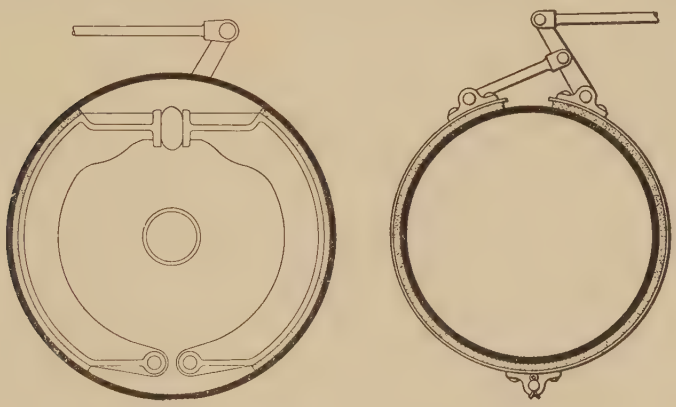


FIG. 1 (LEFT).—DIAGRAM OF INTERNAL SHOE BRAKE.
FIG. 2 (RIGHT).—DIAGRAM OF EXTERNAL BAND BRAKE.

tionary part of a band brake consists of a flexible steel band, generally lined with a woven or molded friction material, which may be brought into contact with the drum by exerting a pull on its ends when the brake is external, and pushing against its ends when the brake is internal. The band is supported at or near its middle by a bracket or plate and an anchorage pin; this structure—which is secured to the axle housing in the case of a brake acting directly on the road wheels—takes up the turning moment due to the friction when the brake is applied. The drum is always the movable or rotatable member of the brake.

The chief difference between a shoe brake and a band brake is that in the former the stationary members are com-

paratively rigid while in the latter they are flexible. Usually there are two shoes inside the drum, each forming an arc of somewhat less than 180°. Each shoe is pivotally mounted on a bracket at one of its ends, the anchorage pins for both shoes being either identical or else located close together. The other, free ends of the shoes can be forced apart by some suitable mechanism, and the shoes thus brought into contact with the interior surface of the drum. Simple diagrams of internal and external brakes are shown in Figs. 1 and 2.

Historical Development—The earlier automobiles were braked by means of their rear wheels only. With the moderate speeds of these vehicles, and the low density of traffic on the roads at that time, braking on two wheels was quite sufficient for safety, and the rear wheels were chosen because it was easier to apply brakes to them, since their planes of rotation are fixed with relation to the axle, and also because usually more than half of the total weight of the vehicle and load is carried on these wheels.

Of the two braking systems, either one or both had drums carried by the rear wheels. Sometimes both brakes acted on the same drums, two internal brakes being located side by side, or else one internal and one external brake being used. One objection that has been raised to this arrangement, and which applies particularly where one brake is internal and the other external, is that it does not enable the operator to keep his brakes cool during long descents by using the two alternately. The objection is of less importance in connection with passenger cars of the lighter type, in which it is usually not difficult to provide brakes of sufficient area to prevent them from overheating. It then has the advantage of simplicity and low cost of construction. Two rear-wheel brakes side by side inside the same drums have been very popular in England in recent years. A further alternative consists in placing four shoes inside a single drum, each subtending slightly less than 90° of arc, and operating opposite ones together as one brake.

Instead of having two brakes act on the same drum, each rear wheel may be fitted with two drums of different diameters, the service brake then preferably being applied to the larger drum. If only one brake acts on the wheels directly, the other usually acts on the transmission drive shaft, such a brake being known as a transmission brake. Finally, a brake drum may be applied to the pinion shaft or worm shaft of the rear axle drive. In the case of worm drive the drum may be located either behind or in front of the axle, whereas with bevel gear drive it can be located only in front

thereof. Such brakes on the pinion- or worm-shaft are comparatively rare, however.

Number of Brakes—The force which causes retardation of the vehicle ordinarily is proportional to the pressure with which the brake bands or shoes are applied against the drums, and to the coefficient of friction between lining and drum. If the force of application exceeds a certain value, the road wheels are locked (held from rotation). Thus the maximum braking effect which can be produced depends upon the adherence between the tire and the road, which in turn depends upon the load carried by the wheels on which the brakes act and upon the coefficient of friction between tires and road.

As the speed of motor vehicles and traffic congestion increased, it became evident that braking by means of the rear wheels alone was no longer adequate. Considerable interest was shown in four-wheel brakes in England for a period of several years starting in 1908, but some of the front brakes then applied proved faulty, and the movement toward the use of four-wheel brakes received a temporary check. It was taken up again in France immediately after the War, and several years later four-wheel brakes were introduced in this country and soon became very popular. One of the principal pioneers of four-wheel brakes was Henrí Perrot, a Frenchman, who as engineer of the Argyll Company of Glasgow, Scotland, in 1909 developed a system of four-wheel brakes that later on, in somewhat modified form, came into wide use.

Calculation of Braking Power—The brakes are generally so designed that it is within the power of the operator to lock the wheels on dry road surfaces. Assuming that six-tenths of the total weight W of the car and load rests on the rear wheels, and that the ground adhesion or coefficient of friction between tires and road surface is 0.6, the braking force when the rear wheels are locked is

$$0.6 \times 0.6W = 0.36W.$$

Suppose that the car is traveling at a speed of V m.p.h. ($= 1.466V$ ft. p. s.). Then the kinetic energy stored up in it is

$$\frac{W(1.466V)^2}{2g} = \frac{WV^2}{30} \text{ (appr.)}.$$

Now let the brakes be applied so as to lock the wheels and let the car come to a stop after running a distance x. Then

$$0.36Wx = \frac{WV^2}{30}$$

and

$$x = \frac{V^2}{10.8}.$$

This equation gives approximately the minimum distance in which the car can be stopped if six-tenths of the total load rests on the wheels through which the brakes act and the ground adherence is 0.6. In the above we neglected to take account of the fact that the air resistance and the rolling resistance encountered by the front wheels also tend to stop the car, acting in the same direction as the friction between tire and ground. It will be shown, moreover, that the car can be stopped in a shorter distance if the wheels are not quite locked. For these reasons the equation for minimum stopping distance would be approximate only even if the assumptions regarding load distribution and ground adherence were absolutely correct. That the equation is substantially correct is shown by the fact that in brake tests made at the Bureau of Standards in Washington in 1924 the average maximum deceleration of four cars with two-wheel brakes on dry asphalt pavement was 11.1 ft. p. s. p. s. while the equation given above implies a deceleration of 11.6 ft. p. s. p. s.

Stopping Distances—For a car with four-wheel brakes, in which the adherence due to the total weight of car and load is available for braking purposes, the minimum stopping distance for a ground-adherence coefficient of 0.6 is given by the equation

$$x = \frac{V^2}{18}.$$

The tests at Washington already referred to showed ten four-wheel-braked cars to have an average maximum deceleration of 20.9 ft. p. s. p. s. (as compared with the 19.3 ft. p. s. p. s. implied by the above equation). The difference is partly accounted for by the wind resistance and the front-wheel rolling resistance, and may be due in part to a slightly higher ground adherence than 0.6.

The important point to remember is that the distance in which a car can be stopped is proportional to the square of the speed it is traveling at when the brakes are applied. Assuming the results of the Washington tests to be representative of the braking powers of modern cars on dry, hard pavements, the distances in which cars can be stopped by means of the brakes on such pavements from various initial speeds, with two-wheel and four-wheel brakes, respectively, are as given in Table I herewith.

Table I—Minimum Stopping Distances on Dry, Hard Roads

Initial Speed (M.P.H.)	Stopping Distance in Feet With Two-Wheel Brakes	Stopping Distance in Feet With Four-Wheel Brakes
10	9.66	5.13
20	38.6	20.5
30	87.0	46.2
40	154.6	82.1
50	241.5	128.3
60	347.3	184.7

Unfortunately, the friction coefficient varies greatly with the condition of the pavement. Tests have shown that on concrete and asphalt pavements, dry, the friction coefficient varies between 0.4 and 0.75. On clean wet asphalt it is about the same but decreases with an increase in tire load. On asphalt covered with a layer of wet dirt it drops to as low as 0.1.

Conditions Insuring the Quickest Stop—It was found in experiments made by the Westinghouse Air Brake Company that railway car brakes exert the greatest retarding effect when applied with such force that the wheels do not quite lock, but continue to revolve. This same condition undoubtedly exists in connection with automobile brakes. It may be explained on the grounds that the friction of rest is greater than the friction of motion, and that when the wheels become locked the rolling friction of the wheel on the ground and the bearing friction of the axle and propeller shaft cease.

The braking effect certainly is the greatest if the energy dissipated in friction in traveling a unit distance is a maximum. Referring to Fig. 3, let R be the starting resistance of the wheel to slippage on the road (friction of rest) and r the resistance of the wheel to slippage once it has begun (friction of motion). Also, let D be the wheel diameter and d the diameter of the wheel-brake drum. Then the maximum frictional force which can be applied to the brake drum without causing the wheel to slip is $(D/d)(R - a)$, where a is a very small quantity. While the car travels a unit distance the brake drum circumference moves a distance d/D in its rotation and the energy absorbed at the brake surface is

$$\frac{d}{D} \times \frac{D}{d}(R - a) = R - a.$$

In addition to this we have the energy absorbed by the roll-

ing friction at the road contact and the bearing friction. We will denote the sum of these two frictional forces, both referred to the wheel rim, by B, and the energy absorbed by these two resistances while the car moves through unit distance may be also represented by B. Therefore, the total energy absorbed while the car travels a unit distance when the brakes are on but the wheels are not locked is $R + B - a$.

On the other hand when the wheels are locked, after sliping has begun (which, of course, occurs instantly), the only resistance encountered is the sliding friction r of the wheel on the road. The energy absorbed in unit distance due to this friction is also r. Hence we must prove that

$$R + B - a > r.$$

It has been stated already that the friction of rest R is greater

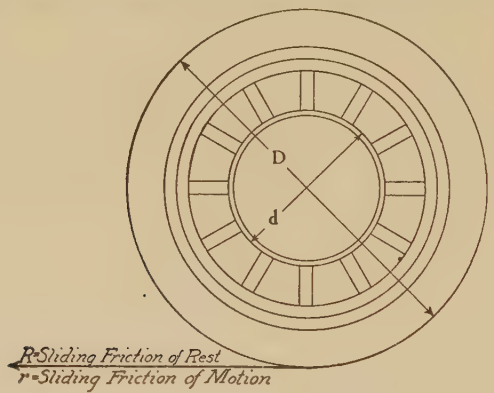


FIG. 3.—MINIMUM-STOPPING-DISTANCE DIAGRAM.

than the friction of motion r. Of the two remaining items B has a definite value which on good roads is from 4 to 5 per cent of the sliding friction. The item a, on the other hand, may be made practically nil, as it represents the margin which, if added to the brake friction referred to the wheel circumference would cause the wheel to lock. Hence, under the most advantageous conditions this item is insignificant and the retarding action is then greater than that due to locked wheels by the sum of the following three items: The rolling friction of the wheels on the ground, the bearing friction in the axle, propeller shaft and transmission, and the difference between the friction of rest and the friction of motion between wheel and road.

Objections to Locking Wheels—Not only will the brakes stop the car more quickly when they are not locked, but the wear and tear on the tires are greatly reduced. Brakes have been designed in which locking is automatically prevented, but they have not come into practical use, probably on account of their complication and excessive weight. With four-wheel brakes the likelihood of the brakes being locked is very much reduced. Locking often occurs when a wheel hits a bump and is thrown into the air, when—if the clutch is out—there is only its rotary momentum to resist locking. In this connection it is interesting to note that the braking effect is increased when balloon tires are used, as with these low-pressure tires the wheel holds the ground much better.

In the foregoing discussion it was assumed that each pair of wheels, front and rear, carried a definite proportion of the total weight of vehicle and load, to which the ground adherence is proportional. As a matter of fact, when the brakes are applied, the forward force of the momentum, acting at the center of gravity of the car, and the rearward force of resistance at the points of ground contact of the wheels on which the brakes act, produce a couple which has the effect of pressing the forward wheels more firmly against the ground and relieving the pressure of the rear wheels on the ground, thus virtually transferring weight from the rear wheels to the front wheels, making the former less and the latter more effective as braking wheels.

Weight Transfer—The transfer of weight from the rear to the front wheels when the brakes are applied to or through the rear wheels may be investigated with the aid of the diagram, Fig. 4. Let W be the total weight of the car, acting vertically downward at the center of gravity, and l_1 and l_2 the horizontal distances of the center of gravity from the centers of ground contact of the front and rear wheels, respectively. Then for the condition of rest the reaction between the front wheels and the ground is

$$r_1 = \frac{Wl_2}{L},$$

L being the wheelbase, and the reaction between the rear wheels and the ground is

$$r_2 = \frac{Wl_1}{L}.$$

Now suppose that the brakes are applied so that the rear wheels slide on the road, with a friction coefficient μ. Then,

neglecting the resistance to the rolling of the front wheels, which is small in comparison with the sliding friction of the rear wheels, the ground reactions will be R_1 and R_2, and there will be a propelling force $R_2\mu$ active at the center point of ground contact of the front wheel,

$$Wl_1 - R_2L - R_2\mu h = 0$$

$$Wl_1 = R_2L + R_2\mu h$$

$$R_2\,(L + \mu h) = Wl_1$$

$$R_2 = W\left(\frac{l_1}{L + \mu h}\right)$$

Also, since

$$R_1 + R_2 = W$$

$$R_1 = W - W\left(\frac{l_1}{L + \mu h}\right) = W\left(\frac{L + \mu h - l_1}{L + \mu h}\right)$$

The amount of weight virtually shifted from the rear to the front wheels by rear-wheel braking therefore is

$$r_2 - R_2 = \frac{Wl_1}{L} - \frac{Wl_1}{L + \mu h} = \frac{Wl_1\mu h}{L^2 + L\mu h}$$

In order to convey an idea of what this really amounts to, we may apply the formulae to a practical case. Suppose we have a car whose loaded weight W is 3600 lbs. and which has a wheel base L of 125 in. Let l_1 be 65 and l_2 60 in., and let the coefficient of adhesion μ be 0.6. Also, let the height of the center of gravity above the road be 30 in. Then the rear-wheel ground reaction r_2 for the condition of rest is:

$$\frac{3600 \times 65}{125} = 1872 \text{ lbs.}$$

Whereas the rear-wheel ground reaction R_2 for the braking condition is

$$3600\,\frac{65}{125 + (0.6 \times 30)} = 1636 \text{ lbs.}$$

Hence the transfer of weight in this particular case amounts to

$$1872 - 1636 = 236 \text{ lbs.},$$

or

$$\frac{236 \times 100}{1872} = 12.6 \text{ per cent.}$$

It will be seen that the lower the center of gravity, the less will be the shifting of weight, and the more effective, therefore, the rear-wheel brakes.

Independent of Brake Location—With the same braking power or retarding force the transfer of weight from the rear to the front wheels is the same whether the braking power is applied through the rear wheels, the front wheels, or all four wheels. There is this difference, however, that with rear-wheel braking the transfer of weight reduces the maximum retarding force which the brakes can produce; with four-wheel brakes the maximum retarding force is not affected by the transfer of weight, while with front-wheel brakes only, the maximum retarding force of the brakes is increased by the transfer of weight. As a corollary to the above, with

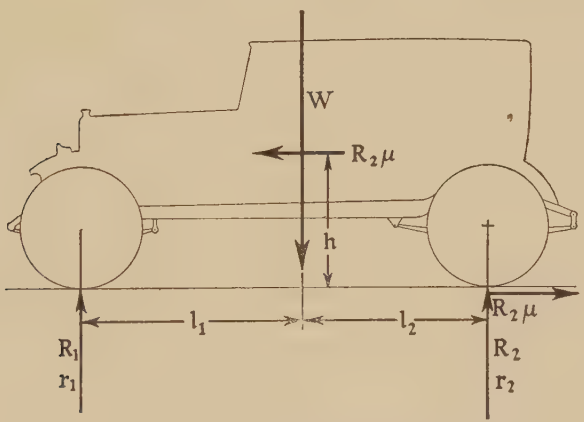


FIG. 4.—WEIGHT-TRANSFER DIAGRAM.

rear-wheel brakes there is no possibility of a car being caused to turn over as the result of powerful brake application, while with front-wheel brakes this is possible on down-grades if the wheelbase is very short and the center of gravity high. This evidently will occur when the weight normally on the rear wheels is wholly transferred to the front wheels, in which case the product μh is equal to l_1. When the car is on a down-grade the height of the center of gravity, h, must be measured above the horizontal through the point of ground contact of the front wheels, and the distance l_1, between the point of ground contact of the front wheels and a vertical through the center of gravity. Thus on a down-grade, h is

increased and l_1 reduced, and there is increased tendency for the car to overturn forward on application of the brakes. Such an accident is impossible with modern low-built vehicles, but it is said to have occurred with front-driven and front-braked electric hansom cabs in service in New York City during the early years of the century.

Brake-Lining Materials—Brakes are generally lined with friction material having a base of asbestos fiber. This material comes in three forms, woven, folded, and molded. In making woven brake lining, yarns or cords of asbestos, etc., are woven into a strong, heavy strip of the required width, which is then impregnated with some hydrocarbon liquid and allowed to dry while under pressure. Folded lining, on the other hand, is made up of asbestos cloth woven of finer yarns, that is treated with a rubber solution. The required number of layers of this cloth are piled on top of one another and are then vulcanized. The yarns used for both types of lining consist of a mixture of asbestos fiber and cotton fiber, the proportion in high-grade linings being about 80 per cent asbestos and 20 per cent cotton, while the poorer grades of lining contain more cotton. Long fiber asbestos and long fiber cotton are best, as they increase the resistance to wear. These fibers are spun around wires of brass or copper.

Asbestos is used in the lining material on account of its ability to withstand high temperatures without injury, and of its comparatively high coefficient of friction. A high coefficient of friction is obviously desirable, because, other things being equal, it reduces the pressure on the pedal or lever required to produce a certain braking effect; but as a rule, with materials of generally the same character, it can be obtained only at the cost of a loss in resistance to wear. The coefficient of friction of any material is dependent upon its hardness and upon its tenacity or toughness. Materials which are soft and at the same time tough, such as natural cork and rubber, have a high coefficient of friction, because their softness enables their surface particles to enter into the microscopical depressions of the metal surface with which they are in contact. Generally, however, any change in the composition which renders the lining softer, also decreases its toughness, and thus its resistance to wear.

Cotton and wire are used in the yarns because they facilitate the manufacturing process. These additions to the asbestos do not change its frictional properties materially, and they make it possible to produce a lining that is more resistant to wear. The cotton content, of course, lowers the ability of the yarn to withstand high temperatures. With

these additions the woven fabric can be made much harder, the wire, for instance, making it possible to twist the yarn more tightly.

Saturants—Treating compounds or saturants are employed to render the fabric substantially impervious to oil and water, and thus to reduce the effects of these liquids on the frictional qualities of the linings. Without the saturant the fabric would absorb a great deal of moisture during wet weather and would then cause trouble either from swelling and seizing of the brakes, or from loss of friction.

Among the saturants or impregnating materials used for brake lining are rubber, asphaltic materials and linseed oil. These impregnating materials are in a certain sense a necessary evil. They are required to prevent undesirable influences of oil and water, but they themselves are the causes of various forms of trouble. Most of these saturants are heat resistant only to a moderate degree; at higher temperatures some of their components become liquefied and ooze out, causing the lining to "bleed." The liquefied portions then act as a lubricant and cause the friction coefficient to decrease. Some saturants will char and produce a dry powder which has a tendency to cause the brakes to grab.

An ideal brake-lining material would have a high and substantially constant friction coefficient, would be free from swelling and from bleeding of the impregnating compound, would be uninfluenced by the presence of oil and water, and would show high resistance to wear.

Friction Coefficients of Linings—The friction coefficient of commercial brake-lining material is not constant but varies with the specific pressure on its surface, the rubbing speed, the temperature and the condition of the surfaces. Rather extensive tests on clutch facings (which are similar in composition to brake linings) were made at the Motor Vehicle Laboratory of the Dresden Technical College by Dr.-Ing. F. Florig. These showed, among other things, that with asbestos fabric against cast iron, the friction of motion is greater than the friction of rest, which is an unusual property. As slippage occurs, the friction coefficient increases, but it soon reaches a maximum value and remains constant thereafter (the temperature being kept constant). The coefficient decreases slightly as the specific pressure increases, and it changes greatly with the temperature, but in a rather irregular way. One of the conclusions drawn from these tests is that, provided oil and water can be kept off the surfaces, a friction coefficient of 0.3 can be depended on with the type of lining mentioned.

Brake-Lining Tests—Extensive tests on brake-lining materials have been made under the direction of S. von Ammon at the Bureau of Standards in Washington, on a special testing machine. The machine is so designed that the load is automatically kept constant in spite of wear of the samples. Tests were run both with water-cooled drums and with ordinary, uncooled drums. In endurance tests the temperature of the lining rises as the test proceeds, although with the water-cooled drum a constant temperature is soon reached.

Such endurance tests on woven linings generally give graphs of the form reproduced in Fig. 5, in which the friction coefficient is plotted against time. It will be seen that at the start of the run there is at first a slight increase in the friction coefficient, from A to B (which confirms Florig's result that the coefficient of motion is greater than the coefficient of rest). During the next phase, from B to C, the friction decreases rapidly to a minimum. This is due to the fact that some of the saturant is liquefied and acts as a lubricant. With the uncooled drum the coefficient of friction remains at the minimum value for a considerable period, evidently until all of the saturant, or the volatile part of it, has been melted or burned out; then it rises again rapidly, from D to E, and thereafter it continues to increase slowly. With the water-cooled drum the maximum temperature of the lining is much lower, and apparently only a small fraction of the saturant is driven off, for after having reached its minimum value the coefficient of friction immediately, or almost immediately, begins to rise again. In the ordinary use of brakes the conditions approach much closer to those of the water-cooled than to those of the uncooled drum, for the brakes generally are applied for brief periods at a time only, and the linings do not reach a very high temperature. The graphs shown in Fig. 5 give a good idea of the influences of temperature on the effectiveness of brakes lined with asbestos fabric.

From a consideration of all of the published data it appears that the coefficient of friction of asbestos fabric against steel or cast iron may reach a maximum value of about 0.65, and may drop to a minimum of about 0.1; a coefficient of 0.3 can safely be figured with if oil and water can be kept off the friction surfaces and the lining can be prevented from reaching excessive temperatures. The greatest weakness of these linings is that their friction coefficient may drop to a very low value when they are overheated. Improvements are constantly being made in brake-lining materials by searching for better impregnating mate-

rials and improving the "heat treatments" applied to make the saturant more heat-resistant. The resistance to wear has been increased materially by subjecting the lining to an intense hydraulic pressure, which compacts and hardens it. Sizes of brake linings have been standardized (see Appendix).

Brake Drums—The drums of wheel brakes for passenger cars in the majority of cases are pressed cold from sheet steel, the steel used necessarily having a low carbon content. The advantage of such pressed-steel drums resides in the fact that when needed in large quantities, they can be produced at little cost. An objection to this material is that, owing to its softness, it is often scored by hard brake lining. Small particles of the steel drum are "picked off" and embed themselves in the lining, and it is these small embedded par-

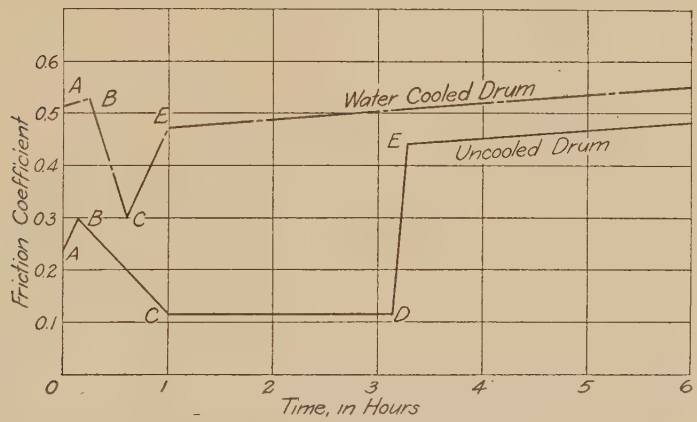


FIG. 5.—CHARACTERISTICS OF WOVEN BRAKE LINING.

ticles that cause the scoring. Another objection to pressed-steel drums is that they are not sufficiently rigid and become deformed under the intense pressure of shoes that are applied with great force, which causes squeaking.

Forged medium carbon steel drums are used to some extent on high-grade vehicles, particularly in Europe, but since these must be machined all over they are comparatively expensive. Where such drums are used they are often formed with cooling ribs on the outside to help radiate the heat. In this country cast-iron and cast-steel drums are used to some extent on heavy vehicles. The cast iron drums have given excellent service, developing a mirror-like polish in service

and showing no trace of scoring. Aluminum alloys have been suggested for use for brake drums, particularly in view of their excellent heat conductivity, but, unfortunately, they have insufficient resistance to wear. This difficulty might be overcome by chill-casting an iron or steel liner in the aluminum drum, but the difference in heat expansion of the two metals is apt to give rise to trouble. Various designs of brake drums with linings of cast iron or similar material are in use, however.

Originally pressed-steel drums were used as they came from the press, without machining, but the demand for increased brake power made it necessary to reduce the clearance between brake and drum in the released position to very small values. This called for greater accuracy of the drums, and they are now generally ground on the braking surfaces, sometimes while on the wheel. Dimensions of pressed-steel drums have been standardized by the S. A. E. (see Appendix).

Drums for transmission brakes of the external type are generally made of cast iron.

Braking Area Required—The amount of energy that must be dissipated in stopping a car is directly proportional to the weight of the car with load, and since the energy-dissipating capacity of a brake increases substantially with its area of frictional contact, the latter should be proportional to the combined weight of vehicle and load. There are, however, a number of other factors to be considered. Since the kinetic energy of the vehicle varies as the square of the speed, more braking surface evidently is required for high-speed than for low-speed vehicles. Vehicles that have to make frequent stops, such as city buses, obviously should have brakes of more liberal size, as otherwise their linings would not last long. Finally, the cooling facilities of the brake and the ability of its lining to withstand high temperatures without injury affect the minimum frictional area required.

From the above it may be judged that any rules regarding braking area required can serve only as a general guide. From the standpoint of effectiveness and long wear of linings, brakes cannot be too large, but in certain types of vehicle, such as high-speed interurban buses with balloon tires, it is difficult to find room for brakes of sufficient size to assure satisfactory wear of the linings.

In passenger car work, four-wheel brakes of the band type are usually made to provide one square inch of friction surface for every 18 lbs. of vehicle and load; internal-shoe-type four-wheel brakes provide one square inch of contact for every 25 lbs. of weight. In moderate-speed motor trucks

with rear-wheel service brakes the braking area may be one square inch for every 60 to 75 lbs. of weight. In high-speed buses four-wheel service brakes should be used, and the contact area made equal to one square inch for about 30 lbs. of total weight. For a city bus the ratio can be made one square inch to 40 lbs. Parking brakes generally provide one square inch of braking surface to every 125 lbs. of car and load.

Detail of Band Brakes—Figure 6 shows an assembly view of an external band-type of wheel brake. The brake drum is merely indicated by dash-dotted lines, the wheel with the drum being supposed to be removed from the axle. The steel band with its lining almost completely surrounds the drum. A lug A is riveted to the band at its middle, by which means

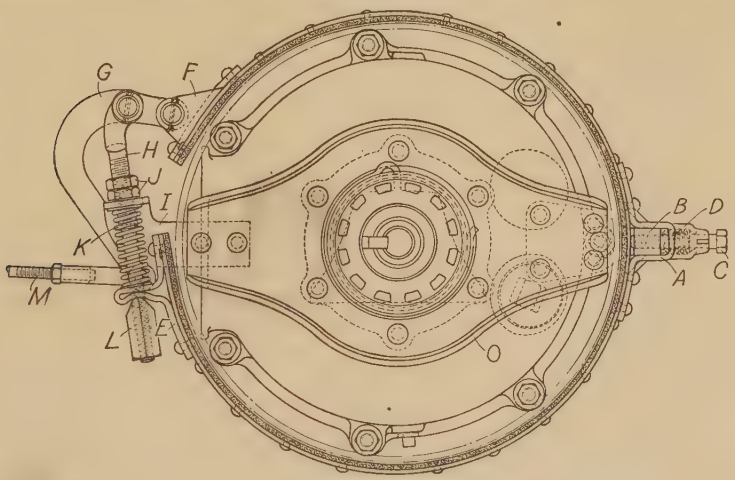


FIG. 6.—TYPICAL CONTRACTING BAND BRAKE (Overland).

it is supported on the anchoring stud B, which in turn is supported by the brake spider O riveted to a flange on the rear axle housing. A cap screw C in the anchoring stud serves as guide for the coiled spring D which withdraws the middle section of the band from the drum when the brake is released. This screw also serves as a stop which limits the clearance of the middle part of the band to a definite amount, which can be varied by adjustment. In this design the anchoring stud B takes the full brake torque and must be of substantial dimensions and rigidly supported.

At its ends the brake band has lugs E and F riveted to it. Lug F has the floating lever G fulcrumed on it. Between

this lever and lug E extends an eye bolt H which passes through a hole in bracket I, the latter being in the form of a pressing riveted to the brake spider O. When the brake is released, bolt H is held in a definite relation to bracket I by means of nuts J and the coiled spring K. Bolt H passes through a hole in lug E and carries a nut L of peculiar form on its end, which serves for adjusting purposes and is self-locking in any position.

It is obvious that when a pull is exerted on lever G through rod M, the band is contracted on the drum, and a braking effect is produced. Spring K serves to withdraw the ends of the band from the drum when the brake is released. Adjustment for clearance at the middle of the band is made by means of screw C, and at the ends of the band by nut L, while nuts J serve to equalize the clearances of the halves of the band.

In many designs of external band brake the contracting mechanism occupies the reverse position to that indicated in Fig. 6, lever G extending up instead of down.

Wrapping Effect—In connection with brake bands the so-called wrapping effect plays an important part. Some of the earlier automobiles were fitted with band brakes in which one end of the band was anchored to the axle housing and the band was applied to the drum by pulling on its other end through a brake rod in a direction tangential to the drum. It was found that such a brake was very effective in stopping forward motion, but had practically no power when the car was running backward. The reason for this is that with forward motion the friction between drum and band tends to wrap the band tighter on the drum, thus assisting the operator in applying the brake, while with reverse motion the same friction works in a direction opposite to the force exerted by the operator.

In Fig. 7 is shown a sketch of a section of a brake drum and band, in which $d\theta$ represents a small arc of contact of the band on the drum and θ the angle or arc of contact between this section $d\theta$ and the point of contact between band and drum nearest to the free end of the band. At the free end a pull P_1 is exerted on the band. Owing to the friction between the band and drum the pull on the band varies from point to point of its length. Let the pull at one side of the differential section $d\theta$ be represented by P and that on the other side by $P + dP$, as indicated in the sketch. Also let the normal pressure between the band and drum on the section $d\theta$ be represented by N and the frictional force resulting therefrom by fN. When the system is in equilibrium the

forces in any direction are equal to zero. Hence, taking the forces in the horizontal plane,

$$(P + dP) \cos \frac{d\theta}{2} - fN - P \cos \frac{d\theta}{2} = 0.$$

But the cosine of an infinitely small angle is equal to unity, hence

$$dP = fN.$$

Now, taking the forces in the vertical plane,

$$N - (P + dP) \sin \frac{d\theta}{2} - P \sin \frac{d\theta}{2} = 0.$$

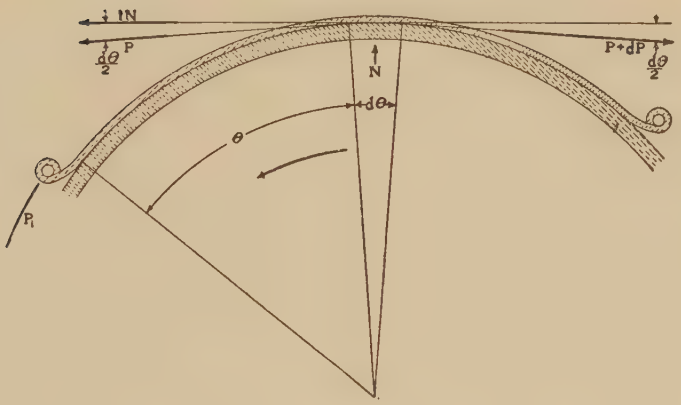


FIG. 7.—DIAGRAM ILLUSTRATING WRAPPING EFFECT.

and since the sine of an infinitely small angle is equal to the arc, we may write

$$N - (P + dP) \frac{d\theta}{2} - P \frac{d\theta}{2} = 0.$$

$$N = Pd\theta + dP \frac{d\theta}{2}.$$

The term $dP\ d\theta/2$, a differential expression of the second order, may be neglected, and we may write

$$N = Pd\theta \text{ or } P = \frac{N}{d\theta}.$$

Dividing the equation for dP by the equation for P we get

$$\frac{dP}{P} = f d\theta.$$

Now the integral of a differential expression of the form dx/x is log x (the natural logarithm, whose base is 2.71828. Hence

$$\log P = f\theta + C$$

or

$$\log P = f\theta + \log c$$

Remembering that in all logarithmic systems the logarithm of the base is 1, we may write

$$\log e^{f\theta} = f\theta \times 1 = f\theta.$$

Inserting this value of $f\theta$ in the equation for log P we have

$$\log P = \log e^{f\theta} + \log c$$

and taking antilogs—

$$P = ce^{f\theta}.$$

To find the value of the constant c we make θ equal to zero, in which case P equals the initial pull P_1 applied to the free end of the band.

$$P_1 = ce^{o}.$$

But any term with the exponent zero is equal to unity, hence

$$c = P_1$$

and inserting this value in the equation for P we have

$$P = P_1 e^{f\theta}.$$

This latter equation gives the pull on the band at any angle θ from the point of contact between band and drum nearest the point of application of the initial pull. The total frictional force F between the band and drum is equal to the difference between the initial pull and the pull at the point of contact between band and drum farthest from the point of application of the initial pull

$$F = P_1 e^{f\theta} - P_1 = P_1 (e^{f\theta} - 1)$$

and

$$P_1 = \frac{F}{e^{f\theta} - 1}.$$

In using this equation, the arc θ must be expressed in radians. Values of the expression $e^{f\theta} - 1$ for various values of $f\theta$ may be found from Fig. 8.

Distribution of Band Tension and Friction—In a band brake with a contracting mechanism similar to that shown in Fig. 6 the pull at both ends of the band must be substantially the same. If P be the pull on each end and P_1 the pull on the anchor pin by the upper portion of the band,

$$P_1 = Pe^{f\theta}.$$

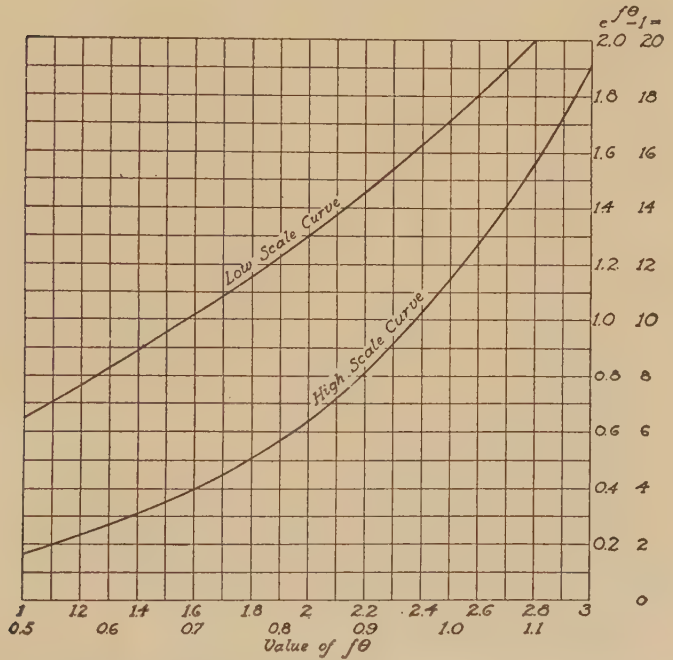


FIG. 8.—GRAPH FOR USE IN CALCULATING FRICTION BETWEEN BAND AND DRUM.

The friction between band and drum between these two points is

$$F_1 = P_1 - P = P(e^{f\theta} - 1).$$

For the lower portion of the band we have the reverse condition. Let P_2 be the pull of this portion on the anchor pin; then, since the pull on the free end is P, we have

$$P = P_2 e^{f\theta};$$

hence

$$P_2 = \frac{P}{e^{f\theta}}.$$

The friction between this portion of the band and the drum therefore is

$$F_2 = P - P_2 = P\left(1 - \frac{1}{e^{f\theta}}\right).$$

With a friction coefficient of 0.4 between band and drum and an arc of contact for the half-band equal to 144° (= 2

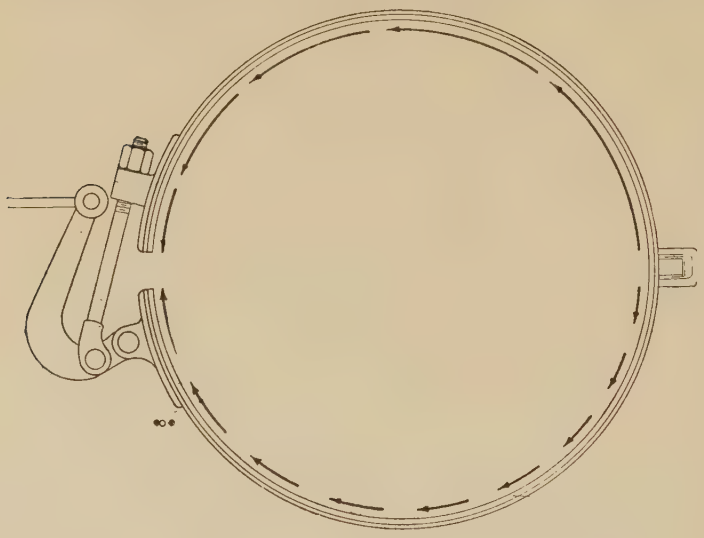


FIG. 9.—SHOWING VARIATION OF TENSION ON BAND.

radians), the value of the exponent $f\theta$ becomes 1.00, and the friction between the top half (or the wrapping half) of the band and the drum is

$$P\,(2.71828^{1.00} - 1) = 1.72P.$$

Similarly, we find the friction between the bottom half of the band and the drum to be

$$P\left(1 - \frac{1}{2.71828^{1.00}}\right) = 0.412P.$$

that is to say, the friction between the top half of the band and the drum is equal to 1.72 times the pull on the free end, and the friction between the lower half of the band and the drum, equal to 0.412 times the pull on the free end. The total friction between band and drum therefore is equal to 2.13 times the pull on one of the ends. This friction is the same whether the drum is revolving forward or backward, because in both cases one-half of the band tends to wrap and the other half to unwrap.

The figures here given are by way of illustration only, and the results would vary with the friction coefficient and with the arc of contact. The tension in the brake band varies somewhat as illustrated in Fig. 9.

Wrapping Band Brakes—The wrapping principle is made use of in band brakes to make application of the brakes in stopping forward motion easier, or, in other words, to produce a sort of boosting effect. For instance, that section of the brake which tends to wrap up when checking forward motion may be made to comprise five-eights of the whole length, and the section which tends to unwrap, three-eights. Then, using the same value for the friction coefficient μ and the total angle of contact as in the previous example, the friction between the wrapping section and the drum will be

$$P\,(2.71828^{1.256} - 1) = 2.338P.$$

and the friction between the unwrapping section and the drum,

$$P\left(1 - \frac{1}{2.71828^{0.754}}\right) = 0.529P.$$

making the total friction,

$$(2.338P + 0.529P) = 2.867P.$$

which compares with 2.13 P for an otherwise similar brake in which both sections of the band are of equal length. Thus in this particular case the off-center anchorage of the brake band increases the brake force for a certain pull on the band by 34.5 per cent, and, conversely, for a certain braking force the pull on the band (pedal pressure) can be reduced to 25.3 per cent. This again is only an illustration, the gain varying with the coefficient of friction and the angle of contact.

Expanding Band Brakes—An expanding band brake is shown in Fig. 10. The band extends over slightly less than 300° of the circumference and has pressed-steel extensions riveted to it, the ends of which have slots cut in them. Into

these slots engage the arms of what is essentially a double-armed lever, which is fulcrumed in the backing plate. The ends of the band are being drawn toward each other by a release spring.

By referring to Fig. 10 it will be seen that there are three eccentrics supported on the backing plate, which serve as stops for the band when released and tend to equalize the clearance all around. In addition to the spring between the ends of the band, there is another one near the middle, which draws that portion of the band radially inward. The brake is anchored through a link between a lug riveted to the inside of the band and a lug secured to the backing plate. It will be noticed that the point of anchorage is off-center, and the

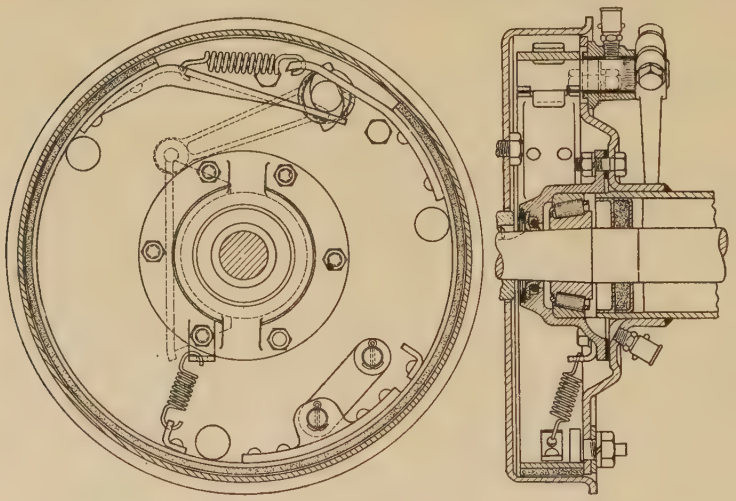


FIG. 10.—TYPICAL EXPANDING BAND BRAKE (Erskine).

brake is designed to give a wrapping effect for forward motion.

Toggle Expanding Mechanism—In designing an expanding mechanism for a band brake it is essential to lay it out in such a way that the pressure against the ends of the band will be nearly tangential, as otherwise the distribution of pressure over the brake surface will be very non-uniform. Toggle mechanisms have been extensively used for expanding band brakes, and such a mechanism is illustrated in Fig. 11. The ends of the band are connected by two pairs of links forming a toggle, and another link presses against the middle

of the toggle radially outward. In order to allow for wear of the lining, the toggle must be so designed that when the lining is new, the links make a considerable angle with each other even when the brake is applied. In that case there is an intense localized pressure near the ends of the band, which is bound to cause the lining to wear rapidly at the ends. To overcome this objectionable feature, toggle expanding mechanisms similar to the one illustrated in Fig. 12 have been used more recently. Here the center of the toggle is pulled inwardly when the brake is applied, and by far the larger component of the force acting along the toggle links is tangential to the band.

Both of the toggle mechanisms illustrated connect by a long link to a crank on the opposite side of the axle. Sometimes the entire mechanism is located on the same side of the axle, but in that case things are rather crowded.

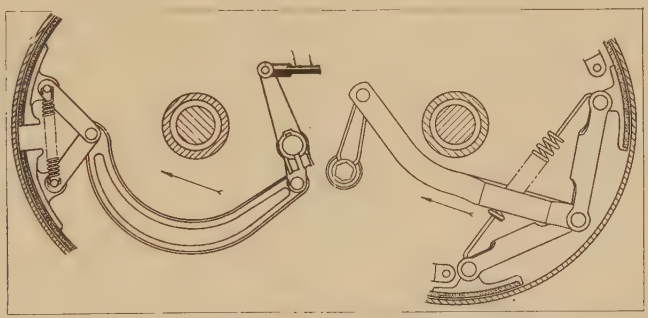


FIG. 11 (LEFT).—TOGGLE-TYPE EXPANDING MECHANISM WITH OUTWARD PRESSURE.

FIG. 12 (RIGHT).—TOGGLE-TYPE EXPANDING MECHANISM WITH INWARD PULL.

The most common expanding device is the cam, and this will be dealt with in connection with shoe brakes. A wedge with a radial motion and a double-armed lever or double crank with link connections to the ends of the band have been used also.

Advantages and Disadvantages of Band Brakes—Among the advantages of band brakes are that they are cheap to manufacture; that they are easily adjusted, as the adjustments are usually on the outside and can be easily reached; that a relatively large fraction of the total cylindrical surface can be made effective as braking area, and that the

lining wears more evenly and therefore is used more economically.

Among the disadvantages of the usual type of external band brakes may be mentioned that they are directly exposed to mud and water, and the frictional action therefore is less constant (although they are better protected against oil from the axle); and that they are more likely to become noisy with wear. With all band brakes it is somewhat more difficult to prevent dragging of the brake when released, for the reason that, owing to the flexibility of the band, the release movement of the whole band is not positive. On account of the nature of the release movement, a greater clearance must be allowed, and the mechanical advantage obtainable in the brake

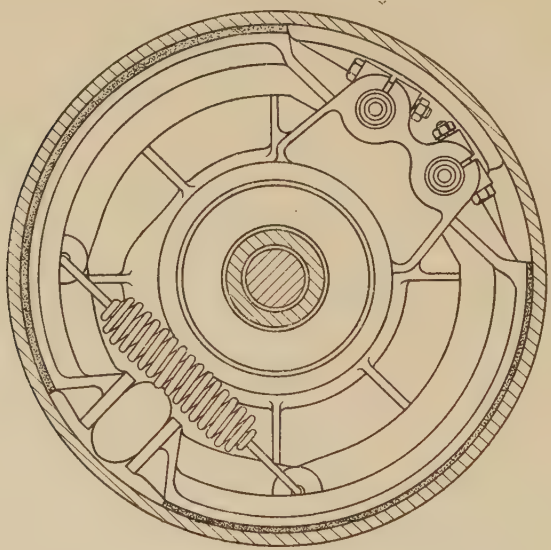


Fig. 13.—Internal-Shoe-Type of Brake.

rigging is less. This latter disadvantage is offset by the ease with which a self-energizing effect can be secured.

External brakes do not get rid of the heat generated so well, since the brake lining is a poor conductor of heat.

Details of Internal Shoe Brakes—Figure 13 shows a brake of the internal shoe type. Shoes for such brakes are made of malleable iron, cast steel, drop-forged steel, pressed steel and aluminum. Malleable iron shoes have been largely used in the past. Pressed steel has the advantage of cheapness

in large-quantity production. Cast aluminum is sometimes used in the more expensive cars, partly because it tends to reduce the unsprung weight, but mainly for the following reason: When the brakes are applied hard in descending long grades, the drums heat more than the shoes, and they "expand away" from the latter, thus reducing the "pedal reserve." If drum and shoe heated at the same rate this would not occur, but the drum always heats more. Now aluminum, having a higher coefficient of heat expansion than the ferrous metals, minimizes this differential expansion, and thus maintains the pedal reserve. The shoes are always made of a section ensuring a high degree of rigidity.

In the drawing the shoes are shown to be supported by a separate anchor stud each. In most designs only a single stud is used. This necessitates either reducing the bearing length of each shoe to one-half, producing a sort of hinge joint, or else cutting away a little more than half of the bearing on each shoe, so that it can only push against its anchorage and not pull on it.

Brake Shoes Always Push Against Anchor—On casual consideration it might seem that when both shoes are fully anchored, the friction on them would cause the self-energized shoe to push against its anchor and the other shoe to pull on it. Since a shoe with half of its eye (or bearing) cut away cannot pull on its anchor stud, the action of brakes with such an anchorage would seem to be different from that of brakes in which each shoe has an eye completely surrounding its anchor. Such is not the case, however. If the shoe which is urged away from the anchoring stud were not pressing against that stud, it could not press against the drum, and there would then be no friction. The pressure of the cam on the shoe always is greater than the force of friction on it, and both shoes press against the anchor stud when the brake is applied. It is therefore preferable, if only one anchor stud is to be used, to cut away a little more than half of the eye on each shoe, rather than to reduce the bearing length of each shoe to one-half. When the brake is applied the shoes are forced against the stud by the force of the cam, and when it is released, they are held there by a coiled spring between them (see Fig. 37).

At the opposite end from the anchoring eye the shoes are usually formed with a flat, hardened surface on which the expander cam acts. With some forms of cam rollers or round buttons must be used. The cams are given a variety of shapes, viz., square or rectangular with rounded corners, oval, or a special profile, the cam outlines being spirals.

Square and rectangular cams have the advantage of giving a rapid initial motion to the shoes and then, when the linings approach the drum, slowing down the motion by increasing the velocity ratio and thus giving greater braking power for a certain pedal pressure. The spiral form of cam gives a uniform or nearly uniform velocity ratio for all positions, and has the advantage that as the lining wears down, adjustment to compensate for the wear can be made by changing the angular position of the lever on the camshaft, without changing the original reduction ratio.

Self-Energizing Effect—There is a self-energizing effect in shoe brakes the same as in band brakes, and the friction between the lining and the drum tends to make one of the shoes grip the drum more firmly and to release the other one. This may be seen by reference to Fig. 14 in which the various forces and reactions on the shoes are indicated. When the system is in equilibrium the moments around any point vanish. Taking moments around the anchor pins, the condition of equilibrium for the upper shoe is

$$Cc = Aa - Bb,$$

and for the lower shoe

$$Cc = A'a + B'b.$$

(We are assuming here that the cam floats, so that its pressure on both shoes is the same.) Hence

$$Aa - Bb = A'a + B'b$$

Since $$B = \mu A \text{ and } B' = \mu A',$$

where μ is the coefficient of friction between lining and drum, we may write:

$$Aa - \mu Ab = A'a + \mu A'b,$$

from which it follows that

$$A = \left(\frac{a + \mu b}{a - \mu b}\right) A'$$

and

$$B = \left(\frac{a + \mu b}{a - \mu b}\right) B'.$$

The coefficient of A' and B' in the above is, of course, considerably greater than unity. In fact, with the usual values of a, b, and μ, its value is around 2.

If the pressure of the cam on both shoes is the same, as

we have assumed and as it always is with some forms of actuating mechanism, the shoe which is forced by friction in the direction of the anchorage, in other words, the self-energizing shoe, will hold much more strongly, and the wear on it will be greater. This was confirmed by H. D. Church, Director of Engineering of The White Company, who in an S. A. E. paper on Internal Wheel Brakes stated that he had found that the self-energizing shoe required relining about twice as often as the other.

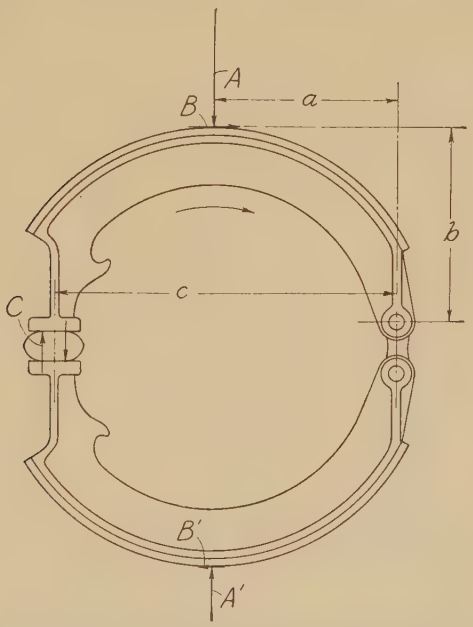


Fig. 14.—Diagram Illustrating Self-Energizing Effect with Shoe Brakes.

Floating Cams—Floating cams are preferred by designers, especially for heavy vehicles, in the case of which pedal pressure must be conserved. There are essentially three methods of mounting the cam in such a way that it will float. The first, which is used chiefly in connection with front-wheel brakes, consists in letting the camshaft overhang its bearing a considerable distance. The bearing is then supported on the axle beam and the camshaft passes through a hole of considerable size in the backing plate, the hole left by the shaft being closed up by a leather boot. The "float" of the

cam in this case is due entirely to flexibility of the shaft and any looseness of the bearing.

The second method consists in mounting the camshaft in a bearing capable of a rocking motion in a plane parallel to the backing plate. This is illustrated in Fig. 15. The third method consists in making the cam in two parts, the cam proper having a sliding motion in a holder formed integral with the shaft. An example of this type of cam is shown in Fig. 32.

Distribution of Pressure and Wear on Brake Shoe—Since the brake shoe can move only in one way, viz., angularly around its fulcrum, when the brake is applied, any point in its frictional surface moves in a direction at right angles to

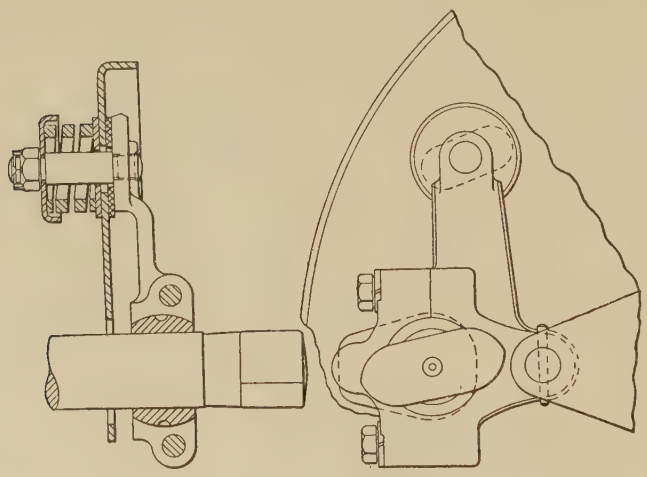


Fig. 15.—Floating Bearing for Brake-Shaft.

a line connecting it with the fulcrum, and at a speed proportional to its distance from the fulcrum. For instance, the point A, Fig. 16, will move in the direction A-B, and we may let AB represent the magnitude of its motion. The motion of point A may be resolved into two components, a radial motion AC and a tangential motion BC.

Now suppose that the lining has barely come into contact with the drum and there is as yet no appreciable pressure between the two. Any further motion can take place only by reason of the elasticity of the materials in contact, those of the lining and the drum. Such further motion will result in the lining being compressed and the drum expanded, and

the compression and expansion at any point of the surface will be proportional to the radial motion BC. Now, in accordance with the principles of elasticity, compression or expansion (extension) is directly proportional to the force causing it, hence the pressure at point A will be directly proportional to the radial motion represented by BC. We can draw similar diagrams of motion for other points of the brake surface, and the resultant of the pressures over the whole friction surface can then be determined graphically as illustrated in Fig. 17. While objections might be raised to the assumption that the resistance to radial displacement of the brake drum is the same at all points, when that displacement is brought about as in this case, the method undoubtedly gives a fair idea of the distribution of pressure, at least under moderate brake applications. The wear of the lining at different parts of the surface will be substantially proportional

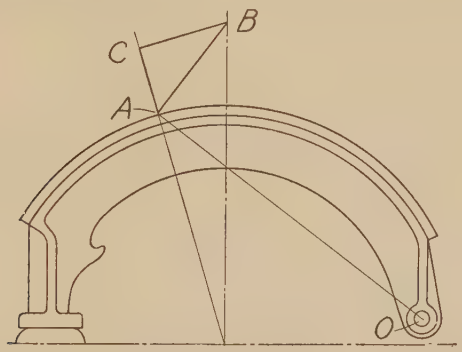


Fig. 16.—Components of Motion of Point on Brake Shoe.

to the pressures at those parts as thus determined. This brake pressure distribution diagram was originated by M. Platt, an English automobile engineer.

Angle Subtended by Lining—For smooth action of the brake (freedom from chatter) it is essential that the lining should not come too close to either the anchoring stud or the cam. Usually an arc of from 30° to 40° on each side of the diameter passing through the anchoring stud and cam axes is left free of lining, and the ends of the lining are skived off.

The brake anchorage studs and their brackets take the entire brake torque, and must be able to withstand this torque without serious deflection. In the lighter vehicles the stud is generally supported unsymmetrically, overhanging its

bracket or the backing plate, but in the heavier vehicles it is usual to employ a symmetrical support which is forked and supports the stud at both ends, or else supports the stud in a single bearing at the middle and the eyes on the brake shoes are forked.

Transmission Brakes—Transmission brakes on passenger cars are usually of the external band type in the United States, and of the internal shoe type in Europe. The design of transmission band brakes does not differ materially from that of band-type wheel brakes. However, the ratio of brake diameter to width of lining is much less than in the case of

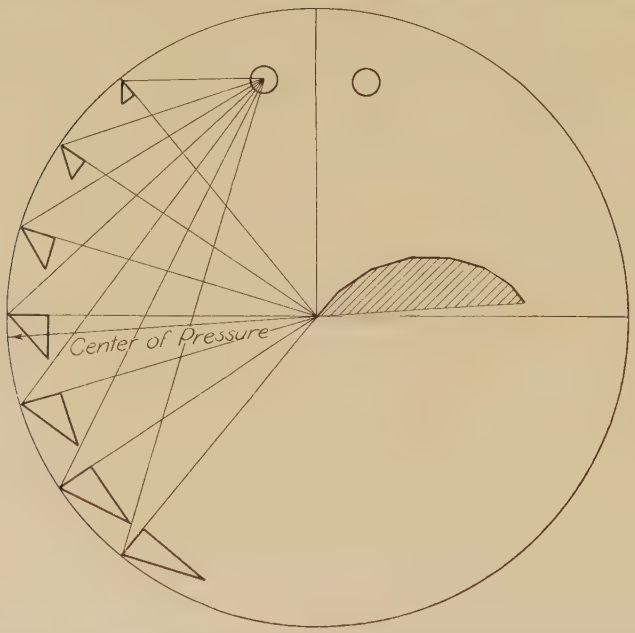


Fig. 17.—Determination of Pressures at Various Points of Brake Shoe.

wheel brakes, the reason for keeping down the diameter of the transmission brake being that otherwise, since it acts through the rear axle reduction and therefore is naturally very powerful, there would be danger of its abuse by careless drivers. Transmission brakes are very easy to operate, but since they are no longer used as service brakes, this is of no great importance. The braking effect of the transmission brake, since it is transmitted to the two rear wheels through

the differential gear, is always automatically equalized. Figure 18 illustrates a band-type of transmission brake and its "hook-up."

With most forms of brake contracting mechanisms a lateral pull is exerted on the brake drum and has to be supported by the transmission rear main bearing. Owing to the fact that the drum, which usually is of considerable width, has a large overhang over the bearing, the load on the bearing due to brake application is very severe, and it is usual to use a bearing one size larger at this point than would be used there if no transmission brake were fitted. The overhang is

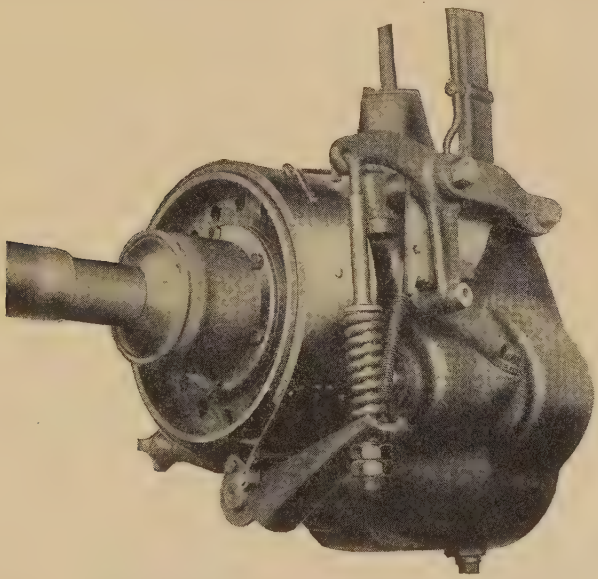


FIG. 18.—BAND-TYPE OF TRANSMISSION BRAKE (G.M.T.).

even greater in motor trucks than in passenger cars, as the brake must be so much larger, and a number of manufacturers of heavy trucks using transmission brakes mount them between two bearings so as to avoid this overhang, or else have a double brake, one-half on each side of a frame cross-member. Fig. 18-A shows a double transmission brake located on opposite sides of a tubular frame cross member, as used on Yellow coaches.

External Shoe Brakes—Brakes with external shoes are used to some extent for heavy commercial vehicles. The shoes

may be either pivoted at one end and connected together at the other by a suitable contracting mechanism, or they may be of what is termed the railroad type, in which the shoes proper are pivoted at their center to swinging arms which are pivoted at one end and connected by a contracting mechanism at the other. A transmission brake of the latter type (as used on the Pierce-Arrow truck) is illustrated in Fig. 19. With the ordinary type of external shoe brake the wear of the lining is faster at the end near the contracting mechanism than at the other, and brake lining therefore is used uneconomically. The railroad-type of brake is

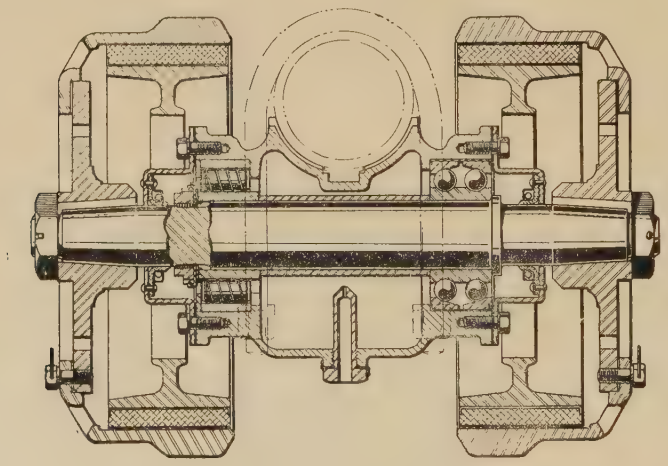


Fig. 18-a.—Expanding-Type Dual Transmission Brake Mounted on Opposite Sides of Frame Cross Member.

evidently an endeavor to remedy this fault, but with it, too, the wear of the lining is non-uniform, being materially greater at the "entering" end, due to the moment of the friction around the fulcrum of the shoe. Attention may be called to the fact that the drum of the brake shown in Fig. 19 is formed with fan-type spokes, for better cooling.

Disc Brakes—Disc brakes have been used to a certain extent both as wheel brakes and transmission brakes. Recently they have achieved a degree of popularity as transmission brakes, because they permit of considerably reducing the overhang over the bearing. A transmission disc brake made by the American Cable Company is illustrated in Fig. 20. The disc is of the double, ventilated type, and there are sector-

shaped, lined shoes on opposite sides of it. Both shoes are suspended by a link from a cross-member of the frame or from a transmission bracket. One shoe is carried by a stud projecting from the link, while the other is mounted on a crank of small throw having a bearing in the link. The crank is provided with a lever arm, and by exerting a pull on this lever the shoes are made to grip the disc. A second pair of shoes may be mounted on the opposite side of the shaft from the pair shown in the drawing, and the capacity of the brake thus increased. By turning the operating lever in the opposite direction from that shown in the drawing, its free end will be in line with the swivel support of the link, and the brake-operating pull then will produce no end thrust on the transmission rear bearing.

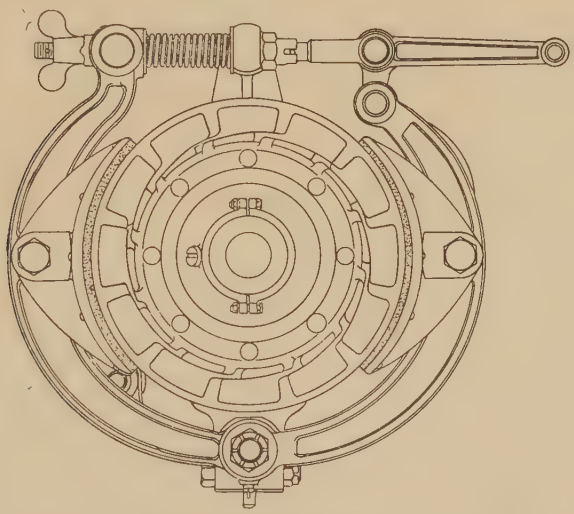


Fig. 19.—Railroad-Type of External Shoe Brake.

Two Brakes Acting on the Same Surface—It always has been considered a disadvantage of the ordinary shoe brake with two shoes (as compared with the band brake) that in order to prevent grabbing and chattering, the area covered by lining must be limited to about two-thirds the available drum area. Provided the distribution of pressure and friction is equally good, a brake naturally will give longer service with one lining the greater the area of its frictional surface. This problem has had due consideration from engineers, and one of the results has been the production of brakes in which

the service shoes and the emergency shoes are placed in the same drums and both occupy the full width of the drums.

A double brake with four equal-sized shoes in each drum, one opposite pair constituting the service brake and the other the emergency brake, has long been supplied on motor truck axles by the Timken-Detroit Axle Co. (Fig. 21). The four shoes together cover nearly the whole of the circumference, and the available drum surface is therefore well utilized. An incidental advantage of this design is that, owing to the small angular length of the individual shoes, the wear on the lining is comparatively uniform. In a Cadillac design two shoes of

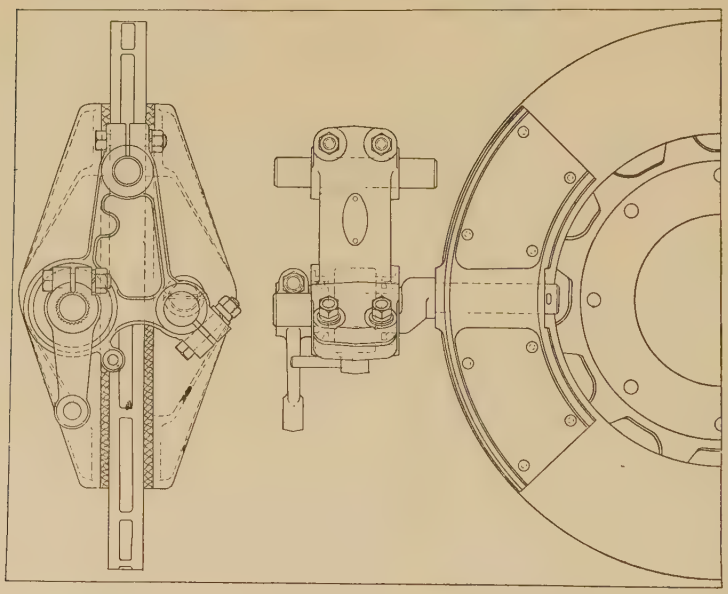


Fig. 20.—Disc-Type of Transmission Brake.

unequal length constitute the service brake and a single, smaller shoe the emergency or parking brake. Of the two service-brake shoes, the larger one is self-energizing for forward motion, and this shoe, instead of being hinged to its anchor pin, is linked to it, to better equalize the wear. The emergency-brake shoe is moved radially outward by means of an eccentric and eccentric rod connected to its middle, the frictional force being taken up by slotted links extending from the ends of the shoe to anchor bolts on the backing plate, one link serving for each direction of motion.

Brake Equalizers—Undoubtedly the ideal method of distributing the force of application between the individual brakes would be to divide it so that the braking effects of the different wheels would be in direct proportion to their ground adherences. This, however, is difficult to accomplish with simple means. Formerly so-called equalizers were almost universally used, by means of which the force of application was more or less equally divided between the brakes.

Four-wheel brakes require three equalizers for their full equalization, one equalizing the force of application between front and rear brakes and the other two equalizing the force

FIG. 21.—TIMKEN FOUR-SHOE DOUBLE BRAKE.

between the two brakes of each set. Alternately, the first may equalize the force between brakes on opposite sides, while the other two equalize the force between the front and rear brakes on both sides. The type of equalizer most used consists of a short balance lever, to the middle of which connects a rod from the brake pedal, while its ends connect through clevises and short links to arms on two cross-shafts (Fig. 22).

An alternate form consists of a portion of a differential gear of the bevel type, a bevel pinion being carried by a lever arm whose free end is connected by a link to the brake pedal. With this pinion mesh two bevel gear sectors that are secured

to the cross-shafts. The whole mechanism can be neatly enclosed, and neatness is probably its chief recommendation.

Cross Shafts Must Be Rigid—One thing that has given a great deal of trouble in brake systems is insufficient torsional rigidity of the cross-shafts. Since the brake pedal is always located to one side of the center line of the chassis, the connection to the cross shafts is also off-center, and one of the shafts is usually about twice as long as the other. If both shafts are of the same diameter the longer one will twist much more under a given torque. Cars have been put out in which it was practically impossible to get any braking effect on the right-wheel brakes (with left-hand control) because the right cross-shaft was so flexible that it would not transmit any appreciable torque. If the brake on that side is set up tighter to get more braking effect, it will drag when released.

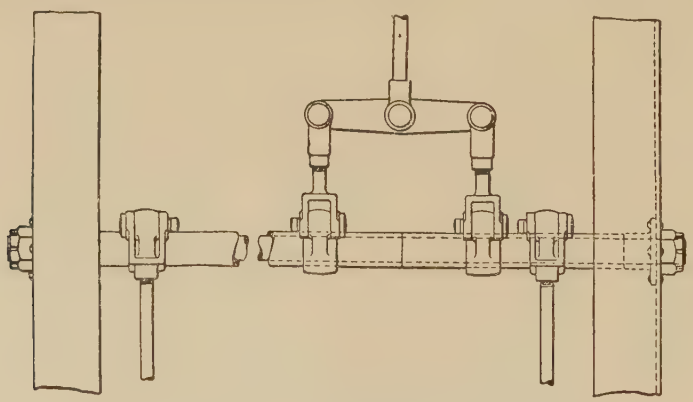


FIG. 22.—BALANCE-LEVER-TYPE OF EQUALIZER.

One European truck manufacturer, in order to obviate trouble of this sort, uses cross-shafts of equal length,, transmitting the motion from the pedal rod to the inner ends of these shafts by means of a hollow shaft concentric with one of them. The simplest means is to use cross-shafts of such large dimensions that there is practically no deflection. A better plan would seem to be to make the two shafts of different diameters, so that although of different lengths, they would flex equally under the same torque load.

Disadvantages of Equalizers—There is considerable difference of opinion among engineers as to whether equalizers are worth while. One objection to them is that they tend to decrease the factor of safety of braking systems. For

instance, in a four-wheel braking system completely equalized, if the connection to one of the brakes fails, the whole system is rendered inoperative. It has been urged against equalizers also that where the connections from them to the different brakes have different degrees of elasticity, the equalizers will not function properly, although that is hardly the fault of the equalizer. Finally, with some systems of front-wheel brakes the principle by which one wheel is released on corners becomes ineffective if an equalizer is used.

Brake Proportioners—In the case of four-wheel brakes the brake force is often divided in unequal proportions be-

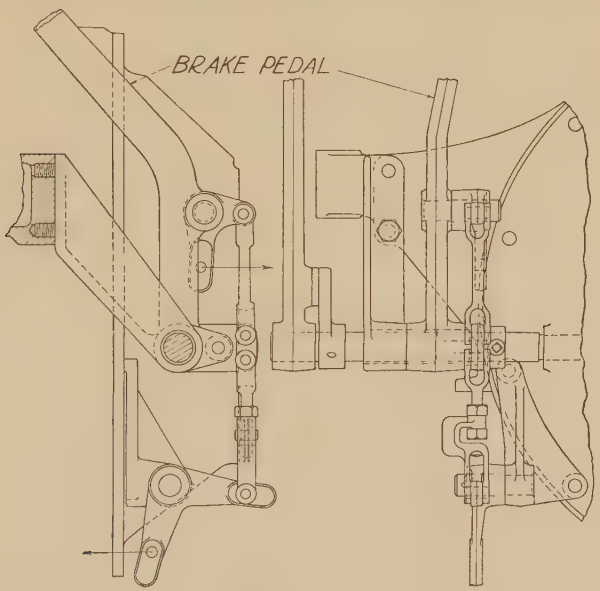


Fig. 23.—Brake Proportioner.

tween the front and rear brakes. A proportion of 60 per cent on the rear brakes to 40 on the front brakes is quite common. The object in reducing the force of application to the front wheels is to make it impossible to lock these wheels, and thus impossible to interfere with the steering function by the application of the brakes. A typical brake proportioner is shown in Fig. 23, the drawing being self-explanatory.

Another scheme for proportioning the force of application between front and rear brakes consists in the use of a cross-shaft carrying double-armed levers at both ends and which

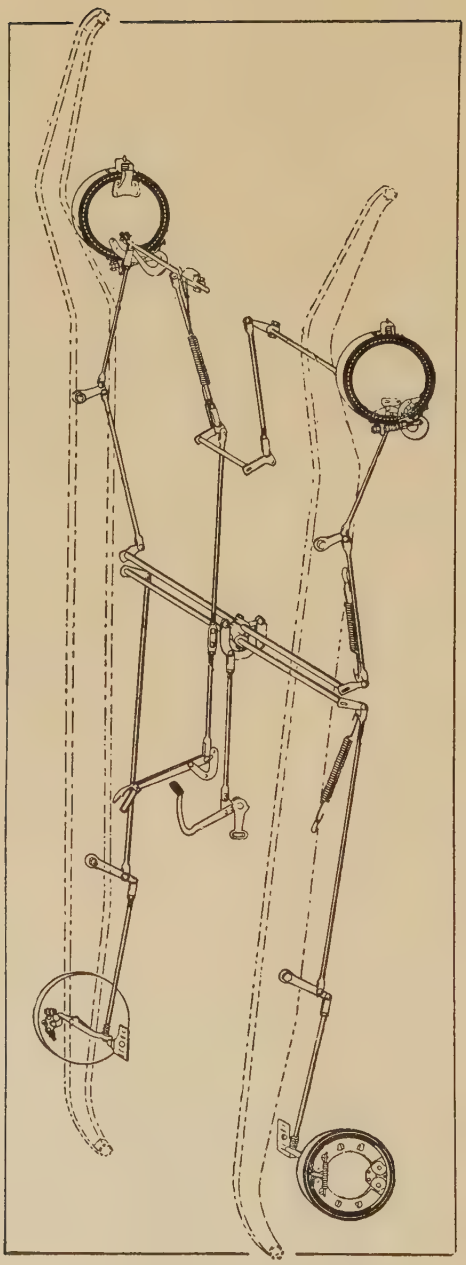


FIG. 24.—TYPICAL "HOOK-UP" FOR INDEPENDENT BRAKES.

is supported in bearings on a frame cross-member at a considerable distance from these levers. From the double-armed levers, rods or cables extend to the brakes on the front and rear wheels. Since the bearing support is comparatively far from the end of the shaft, the end will yield sufficiently to cause equal pulls to be exerted on both rods. Differences in braking effects at front and rear can be secured by making the lever arms or the cams of the two brakes different.

Brake Hook-Ups—There are many different methods of connecting up the pedal and lever with the brakes, but all may be divided into two classes, according to whether a separate brake is provided for emergency use (or for parking), or whether both pedal and lever operate the same brakes. The laws of many states call for two separate brakes, but it has been demonstrated that with certain hook-ups eliminating equalizers, no failure could occur in any part of the mechanism of a four-wheel brake system that would not leave at least two of the brakes effective, and motor vehicle administrators approve or at least raise no objection to such installations.

A typical hook-up for installations comprising separate emergency brakes is illustrated in Fig. 24, this particular hook-up being used on Pontiac and Chevrolet cars. A rod extending rearward from the pedal connects through a yoke-shaped brake proportioner or equalizer to lever arms on two adjacent cross-shafts. These cross-shafts extend to the outside of the frame, and there carry arms from which rods connect to the arms of the brake shafts at both front and rear. In each case the connection is made by two rods in tandem, whose adjacent ends are pivoted to an idler lever fulcrumed on a frame bracket. If connections were made through a single rod, the rods would be too long, and therefore would vibrate and be likely to break. Since the two cross-shafts rotate in opposite directions it is necessary that when the brake is off, the eyes of the levers on them be a certain distance apart, so as to give sufficient range for application.

The emergency brake in this case consists of internal bands acting on the rear wheels. The shafts of these brakes extend parallel with the rear axle and are supported in bearings mounted on the axle housing. At their inner ends they carry lever arms from which rods extend to a short cross-shaft on a frame cross-member, which in turn is connected by a rod directly to the hand lever.

A hook-up for a system in which the lever applies the rear-wheel brakes of a four-wheel system, as used on Autocar trucks, is shown in Fig. 25. The pedal connects by a link

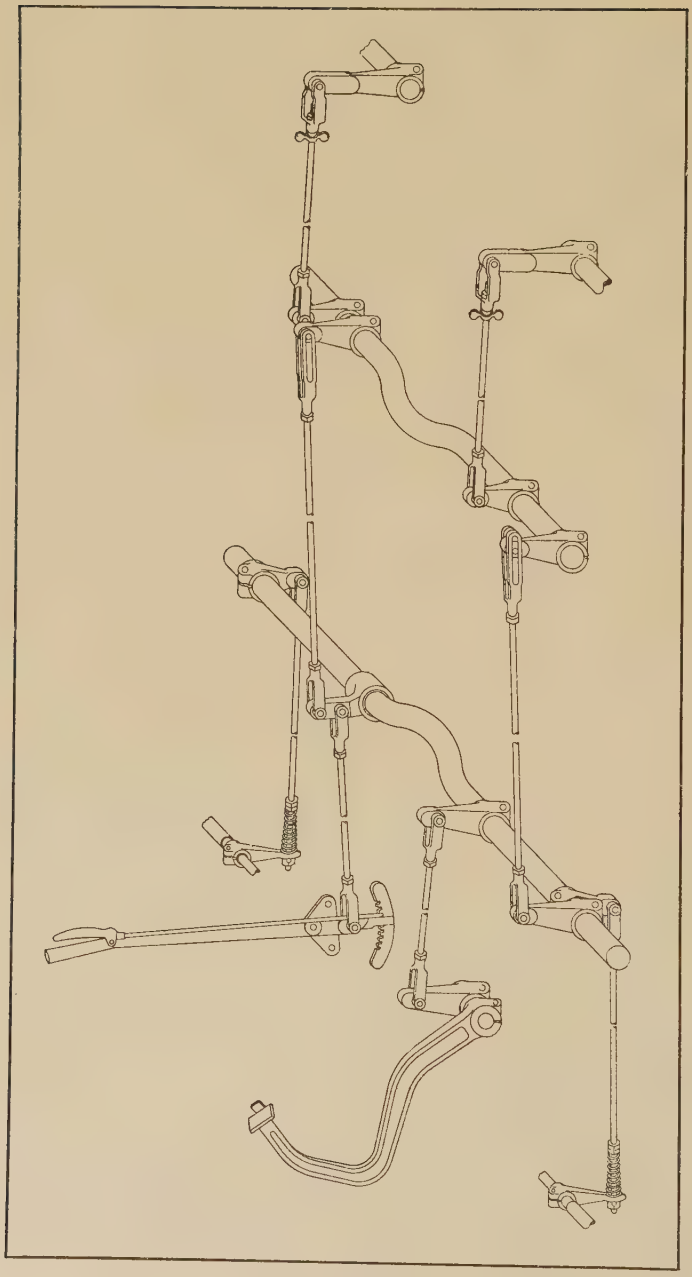


FIG. 25.—TYPICAL "HOOK-UP" FOR FOUR-WHEEL BRAKES WITH BOTH HAND- AND FOOT-OPERATION.

to a forward cross-shaft, which latter connects by another link to a rear cross-shaft, connection being made by means of a slotted connector. From the rear cross-shaft direct link connections are made to the rear brakes. From the brake lever connection is made to the rear cross-shaft through two links in tandem, the connection between the two being by a floating lever on the forward cross-shaft. Connection to the rear cross-shaft from the brake lever also is through a slotted connection. Such connectors are sometimes referred to as over-running joints. Their effect in this case is that when the brakes are applied by the pedal, the brake lever is not moved, while when they are applied by the lever, the pedal is not moved. Moreover, operation of the lever applies only the rear brakes.

Instead of a slotted connector, a jack-knife type of lever is sometimes used for the same purpose. This lever, which is secured to a cross-shaft, becomes rigid when a pull is exerted on its rod, but folds up like a jack-knife when the cross-shaft is turned on its bearing by the other brake-operating device.

As a rule, rods extending to the brake levers of band brakes on the rear wheels are outside the frame, while rods connecting to the brake levers of internal rear-wheel brakes are located inside the frame.

It is very desirable that the entire brake rigging be made substantial and rigid, so that there is very little "give" to it. Not only does this enhance the feeling of security of the driver, but it reduces the effort necessary for brake operation. Effort is required to stretch and twist the parts of the brake rigging, in direct proportion to the stretch and twist, and this is waste.

Brake Cables—Steel cables may be used in place of rods for operating the brakes. One advantage is that a multi-strand cable will not break as a result of vibration. A type of cable introduced in America for brake connections, known as the Trulay, has its individual wires and strands preformed to the shape they assume in the completed cable, so there is no tendency for the cable to uncoil. Fittings are used with this cable which, after being drilled out, are forced down upon the cable by means of a swaging process, which is said to give them a grip that exceeds the tensile strength of the cable itself. The joint, of course, is a much neater one than that obtained by following the old plan of passing the cable through an eye on the lever and fastening its end by clamping.

In order to prevent rattling of parts of the brake linkage when the brakes are "off," a retracting spring is connected

to a lever at one end and a stop is provided for a lever at the opposite end, so that all parts are held taut.

Brake Adjustment—As the lining of the brakes wears, the pedal will travel further forward in applying the brake, and eventually a point would be reached where it would end up against the toe-board without the brakes taking a firm hold. This necessitates some form of adjustment. Brake adjustments must be so designed that they are easy of access without removing a wheel or other important part; they must be easily workable and must be immune to deterioration from rust.

The simplest means of adjustment consists of a device for shortening the brake rod. If one end of the rod is screwed into a yoked connector, the rod can be shortened by loosening

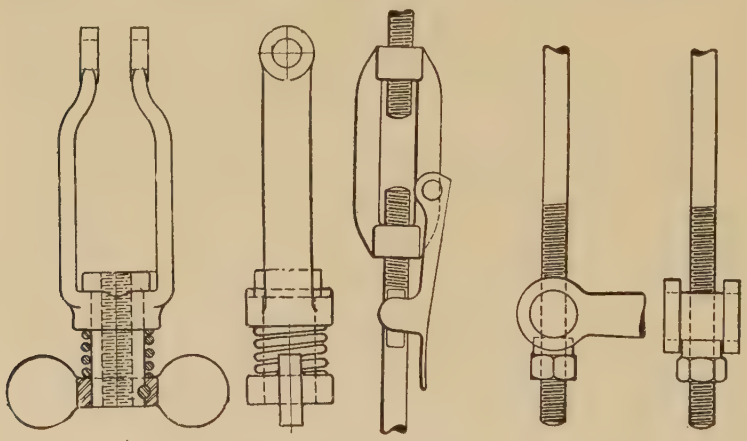


Fig. 26 (left).—Ford Brake-Rod Adjuster.
Fig. 27 (center).—Turnbuckle for Brake Adjustment.
Fig. 28 (right).—Self-Locking Brake-Adjusting Nut.

the connection and screwing the connector farther over the rod. This, however, is somewhat tedious and also is open to the objection that a split pin has to be removed.

A patented device for adjusting the length of the rod without taking it apart is illustrated in Fig. 26. A turnbuckle can also be used, and a special design of turnbuckle with a locking clip, that has been used for this purpose, is shown in Fig. 27. A device for taking up on the rod at the eye of the brake-lever arm is illustrated in Fig. 28. The brake arm takes a trunnion, through a hole in which the

brake rod passes. The nut screwed over the rod is provided with an extension with flat sides, which fits between the arms of the forked connector, thus locking the adjustment. A development of this design is more extensively used at the present time. This consists of a wing-nut with a wedge across its base, this wedge entering a V-slot in the surface on which the nut bears. A coiled spring surrounding the rod forces the wedge into the slot and thus locks the adjustment.

However, shortening the brake rods is not the best method of adjusting the brakes, for as the lining wears down the cam comes into a different angular position, and the leverage is changed. It is therefore customary with internal brakes to use serrated joints for the brake levers and clamp the levers

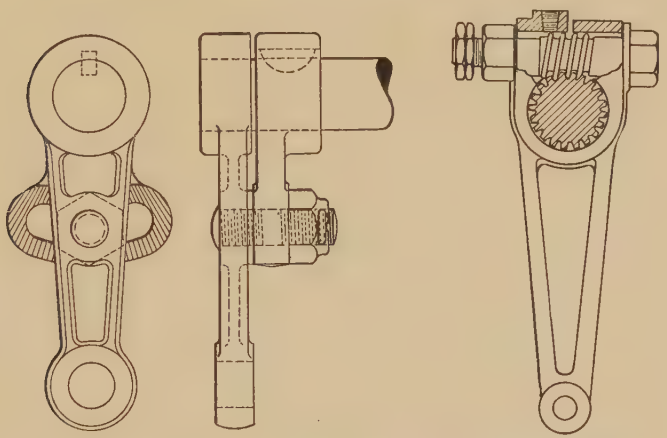


Fig. 29 (left).—Two-Part Lever for Brake Adjustment.
Fig. 30 (right).—Worm-and-Wheel Brake Adjustment.

down on the shafts (see Fig. 35). This permits of making slight angular adjustments of the lever, as may be required. Alternately, the levers may be made in two parts which can be moved angularly with relation to each other and clamped together, the clamping surfaces being serrated, so that friction does not have to be depended upon (Fig. 29). Another scheme, employed in connection with Bendix brakes, consists in cutting the brake shaft with worm-wheel teeth and mounting a worm in the hub of the brake arm (Fig. 30), so that by turning the worm with a wrench, the arm can be moved angularly on the shaft. In an alternate construction a rack

replaces the worm and meshes with spur teeth cut on the shaft. Both ends of the rack are threaded and carry nuts, and by loosening up on one nut and tightening the other, the lever can be moved angularly with respect to the shaft.

The adjusting mechanism illustrated in Fig. 31 is used by Cadillac. A pressed-steel lever is mounted free on the brake camshaft, surrounding a hub which is secured to the shaft by splines. A short length of steel cable extends between a lug on the hub and a spur on the lever, coiling about halfway around the hub. A fitting at the end of the cable carries a nut, and by screwing up on this nut the angular relation of the lever to the shaft can be very finely adjusted.

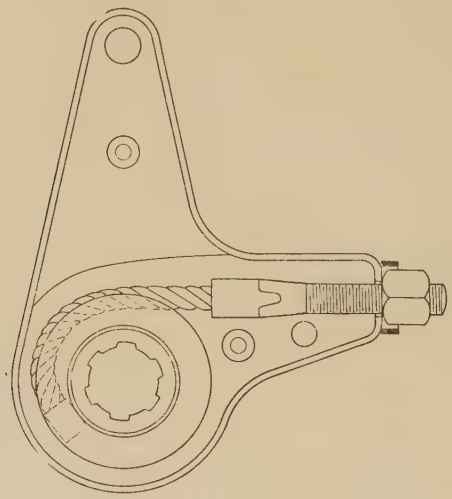


Fig. 31.—Cadillac Brake-Adjusting Mechanism.

Adjustment at Anchor—With internal shoe brakes the lining wears more rapidly near the toe end than near the heel end. This, of course, is undesirable, because it means uneconomical use of expensive lining material. An attempt to remedy this state of affairs is represented by the brake of the Columbia Axle Co. illustrated in Fig. 32. An adjustment is here provided on the anchoring pins. There are two of these pins, and both have a deep slot with inclined bottom machined in them. A steel key of trapezoidal shape, pinned to a headless bolt, is drawn into these slots by the nut on the screw which is located on the outside of the backing plate. This, naturally, forces the anchor pins apart and compensates

for wear of the lining at that end of the shoes. Eccentric anchor pins have been used for the same purpose.

Great importance is attached to accessibility of brake adjustments in Europe, and a good many European cars have shafts carrying small hand wheels projecting through

FIG. 32.—BRAKE WITH ADJUSTMENT AT ANCHORAGE.

the floor-board in the driver's compartment, by means of which brake adjustments can be made even while the car is running. The effect of such adjustments is to shorten the brake rods.

Brake Clearance—When four-wheel brakes came into use it was soon found that their application to the locking point was beyond the strength of the average person, in the case of the heavier vehicles, and a study of the relationship between braking power and pedal pressure was then made. The pressure which must be exerted on the brake pedal to lock all four wheels is given by the equation

$$P = \frac{WfD}{d\mu re} + R,$$

where P is the pedal pressure required, in pounds;

W, the weight of the vehicle with load;

f, the coefficient of ground adherence;

D, the road wheel diameter;

d, the effective diameter of the brake drum;

μ, the coefficient of friction at the brake surface;

r, the ratio between pedal pad movement and the corresponding movement of the brake shoe when practically "on";

e, the mechanical efficiency of the brake rigging;

R, the force of the retractor spring, reduced to pedal pad motion.

Obviously the factors in the numerator of the equation cannot be changed, and in order to make braking easier we must consider the factors in the denominator. An increase in the brake diameter and in the coefficient of friction of the lining will help, but these factors have their limits. The factor in connection with which the greatest improvement was made is r, the ratio of pedal motion to brake shoe (or band) radial motion. This is not constant but varies for every position of the pedal, owing to the use of levers, cams, etc., in the brake rigging. What really counts is the ratio when the brake is in the "on" position. But the average ratio is a fair criterion of this factor.

The total motion of the brake pedal is limited to about 5 in. if its operation is to cause no discomfort. On very heavy vehicles a somewhat greater range of motion may be necessary. Of this 5 in. about 3 in. must be reserved to allow for

wear of the lining and—in the case of internal brakes—expansion of the drums from heating. Allowance must also be made for the elastic yield of the linkage when transmitting the force of application, and this takes up another inch, leaving only one inch of effective pedal pad travel for applying the brake when newly adjusted. With internal brakes the drum heats much more rapidly than the shoes, and the former therefore "expands" away from the latter.

The average clearance of the shoes of internal brakes when first adjusted is about 0.006 in., and it is found in practice that the largest reduction ratio which can be safely figured with is about 150 to 1, which corresponds closely with the figures above given.

In external band brakes the clearance must be made considerably more, owing to the non-positive character of the release. In passenger cars it is customary to specify 0.010-0.015 in. for the clearance at the anchorage, and 0.025-0.030 in. at the free ends. To reduce the pedal pressure required, the brake shafts are sometimes mounted on roller bearings.

Such small clearances as here given can be used only with hard, dense lining materials, which will not absorb moisture to any extent and swell, which would cause dragging. Dragging of external brakes is apt to cause locking, since it will heat and expand the drums.

The fine clearances given for shoe brakes call for grinding of both the drums and the shoes, the latter being ground on suitable fixtures.

Connections to Wheel Brakes—The lay-out of the connections between the unsprung wheel brakes and the sprung members of the brake rigging on the chassis frame offers a problem of considerable importance, since the spring action must not noticeably affect the brake setting. If the brake linkage is improperly laid out, when a truck is loaded or unloaded on a grade with the brake set, the act of loading or unloading and the resulting spring compression or recovery may either release the brake or else set it so tightly that the latched lever cannot be released. The problem consists in finding such locations for the brake-shaft lever and the cross-shaft lever to which it is connected, that the variation in the distance between the eyes of the levers due to spring action will be a minimum.

There are two different cases to be considered, viz., that of vehicles having radius rods, and that of vehicles with Hotchkiss drive. The problem will here be considered as it applies to rear-wheel brakes, the application of the methods outlined to front-wheel brakes being obvious.

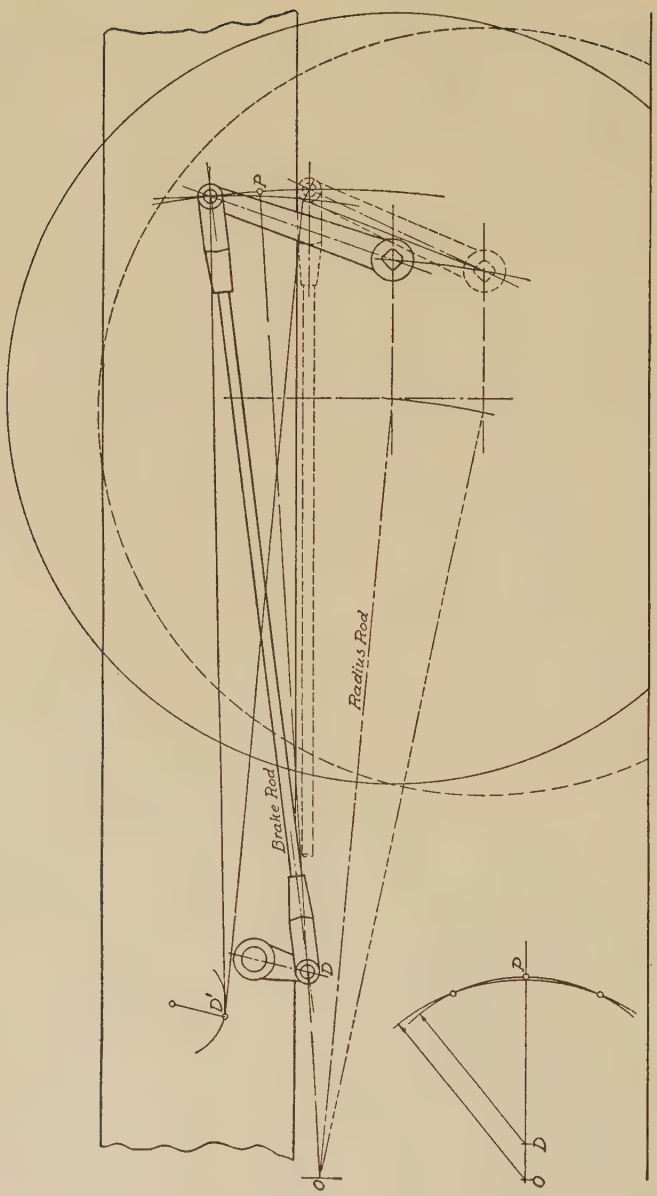


FIG. 33.—LAY-OUT OF REAR BRAKE RODS ON CAR WITH RADIUS RODS.

With a car having radius rods, as the springs compress and recover, the rear axle and every part supported by it move in circular paths around the axis of the forward radius rod connection. Therefore, in order to obviate any influence of spring action on the application of the brakes, the center D (Fig. 33) of the forward brake-rod connection should lie in the axis of the forward radius-rod connection. This, however, is generally impossible in practice. The best practical solution of the problem is to place the forward connection D of the brake rod on the line connecting the axis O of the forward radius-rod connection with the point P representing the mean position of the center of the rear brake-rod connection with relation to the frame. The forward brake-rod connection D may be either forward or to the rear of the forward radius-rod connection O, but should be as close to it as conditions will permit. The sub-figure shows that there is still a slight effect of the spring action on the brake, but with relatively long brake rods it is negligible. When D is located above the line O P (as shown at D′) the brakes tighten when the load is removed, whereas when D is below O P, the brakes loosen.

Brake Layout for Hotchkiss Drive—In trucks employing the Hotchkiss drive (in which the only connection between the rear axle and the frame is by the springs) special problems arise in connection with the layout of the brake-operating mechanism. In this case we must take into account not only the effect on the brakes of the change in relative position of frame and axle when the truck is loaded or unloaded, but also the angular yield of the axle housing under the influence of the braking torque. The angular motion of the axle under the influence of the braking torque is not around the wheel axis as a center, because the chassis spring is not a symmetrical member with respect to torsional loads. The axis around which any movement under the influence of the brakes takes place lies ahead of the wheel axis. If the axis of the eye of the brake lever coincides with this axis of torsional yield, then the application of the brakes does not change the distance between the front and rear supports of the brake rods.

It is readily seen that, looking at the truck from the left side, if the rear brake rod is located above the axis of torsional yield, the distance between the points of brake-rod support will be shortened by the yielding of the axle housing under the braking torque, as this is in a counter-clockwise direction, and if the brake-rod axis is located below the axis of torsional yield, the distance will be lengthened.

Walter C. Baker, who studied the problem in connection

with the Army Class B truck, reached the conclusion that the center of torsional yield cannot be located by graphical means or calculation, and he devised a testing apparatus for locating it experimentally. As illustrated herewith (Fig. 34) the device consists of two readily adjustable attachments, representing the front and rear brake connections; a dial gage and a pair of rods. The device is installed on the chassis as shown, and a turning effort is then applied to the rear axle housing by means of a lever arm and a chain and block. If the front end of the gage rod is fixed, any movement of the rear rod will be indicated by the gage. The rear attachment is then adjusted in a horizontal plane until a position is found which gives the same indicator readings with extreme changes in torque. Next the process is repeated to determine the proper location of the brake-rod rear end in the vertical plane.

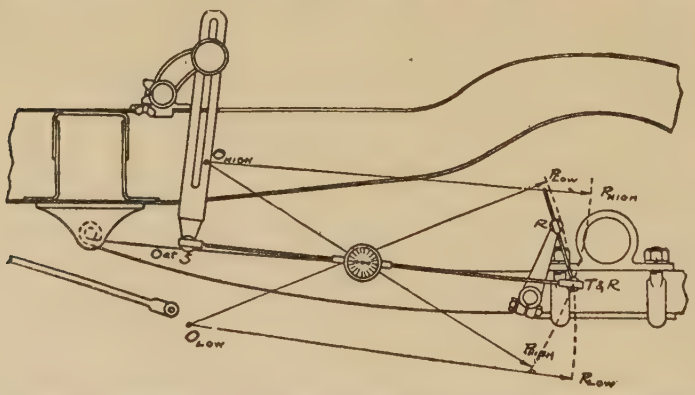


FIG. 34.—DETERMINING PROPER LOCATION FOR BRAKE-LEVER EYES WITH HOTCHKISS DRIVE.

The intersection of the two planes is the axis of torsional yield, and the rear brake-rod center should be made to coincide with it.

If it is impossible to place the rear brake-rod center in the axis of torsional yield, owing to interference, then it should be placed as near it as possible and in line with it and with the correct position of the front brake-rod axis, with a normal load on the springs.

Four-wheel Brakes—Four-wheel brakes are used because they permit of stopping a car from a given speed in a much shorter distance. In most vehicles the rear wheels normally carry only little more weight than the front wheels, and because of the transfer of weight from the rear to the front

wheels by the application of the brakes, the front-wheel brakes are equally effective as, or more effective than, the rear-wheel brakes. The Bureau of Standards in Washington in tests made during March, 1924, found that the average maximum retardation of ten cars fitted with four-wheel brakes was 20.9 ft. p.s.p.s., as compared with 11.1 ft. p.s.p.s. for four cars with two-wheel brakes.

Cause of Skidding—Incidentally, four-wheel brakes make driving safer by lessening the tendency of a car to skid One very common cause of skidding is the locking of the rear wheels by the brakes. Skidding, of course, occurs primarily on greasy roads. As long as the rear wheels revolve, their motion is a rolling one, and they will progress in a direction parallel to their planes. The reason for this is that resistance to rolling motion is many times less than the resistance to sliding motion. However, on greasy roads even a slight application of the brakes is likely to lock the wheels—unless the clutch is left in, as it always should be under such conditions—and once the wheels are locked they will slide as easily sideways as forward. All that is then necessary to start a skid is that one front wheel encounter a resistance slightly greater than that encountered by the other.

A rear-wheel skid cannot be produced by locking the front wheels. When no brakes are applied to them, the rear wheels keep on revolving and act as guides, keeping the vehicle in line. When brakes are applied to all four wheels the likelihood of locking both rear wheels is, of course, greatly reduced; under the same road conditions it will occur only when the braking force is substantially twice as great.

Requirements of Front Brakes—Four-wheel brakes consist essentially of front-wheel brakes added to rear-wheel brakes, and the problems of four-wheel brakes, as contrasted with those of rear-wheel brakes, are thus the problems of front-wheel brakes. Since the front wheels are used for both steering and braking, it is essential that application of the brakes should not interfere with their steering function, and steering should not seriously interfere with the braking function. For instance, when the brakes are released, deflection of the wheels even to the extreme position must not apply them. Since wheels that are locked are incapable of performing the steering function, it should be practically impossible to lock both front wheels. However, contrary to general belief, locking of the front wheels is not more dangerous than locking of the rear wheels, as the latter will cause rear-wheel skids, while the former will not. Application of the brakes must not make it impossible to swing the wheels

around their knuckle-pin axes for steering purposes, but it also must not tend to swing the front wheels around these axes.

When brakes are applied to the front wheels the steering knuckle pins must be so arranged that their axes produced strike the ground at a point slightly inside the center point of tire contact. It is not desirable to have the axis of the pin produced pass exactly through the center point of tire contact, as that would unnecessarily increase the effort required to deflect the front wheels when the car is standing still. Since with front-wheel brakes the front axle and front springs are subjected to torsional forces in addition to the bending forces due to the loads carried by them, these parts must be specially designed with a view to enabling them to withstand the additional loads.

Types of Front-wheel Brake-operating Mechanisms—In their general design front-wheel brakes are similar to rear-wheel brakes. Both shoe and band brakes may be used on the front wheels, internal and external. The chief difference between front- and rear-wheel brakes is that with the former, owing to the motion of the wheel around the knuckle pivot in steering, some flexible means or "universal" mechanism must be used for transmitting the force of application to the brake. The power for brake operation can be transmitted mechanically, hydraulically, pneumatically and electrically. Mechanical transmission naturally involves some complications, because of the relative motion between the frame (whence the operating power must come) and the brake, and this explains why there has been considerable development of the other systems, particularly the hydraulic one. Partial pneumatic transmission may be considered used in the case of air brakes, where the air diaphragm chambers are located close to the brakes. Electric brakes, in which the operating power is carried directly to the brakes by means of electric cables, are used to a slight extent on trucks. In the following pages most of the prominent systems will be briefly described and illustrated.

Perrot Control Mechanism—Figure 35 shows part of the operating mechanism of the Bendix-Perrot front-wheel brake. The brake is of the internal shoe type and is cam-expanded. The camshaft, which is supported in a bearing mounted on the backing plate, connects through a universal joint with a shaft extending to the car frame, on which latter the shaft has a ball-and-socket support. The center of the universal joint is located in the axis of the knuckle pin. From a lever arm, which has a serrated, clamped connection with the shaft

just outside the frame, a rod runs toward the rear and connects with a lever on a cross-shaft to which the brake pedal is connected by a linkage.

A feature of the Perrot system is that it inherently prevents locking by the brake of the outside front wheel when turning corners, thus making it impossible to lose steering control. This results from the use of a universal joint of the type having two axes at right angles, in the connection to each brake, together with the absence of an equalizing device in the connections to the front brakes.

Automatic Release of Outside Wheel—In the chapter on Universal Joints it was pointed out that when the shafts connected by such a joint make an angle with each other, the motion is not transmitted uniformly; that during a quarter of a turn the driven shaft will turn faster than the driving shaft, and during the next quarter it will turn more

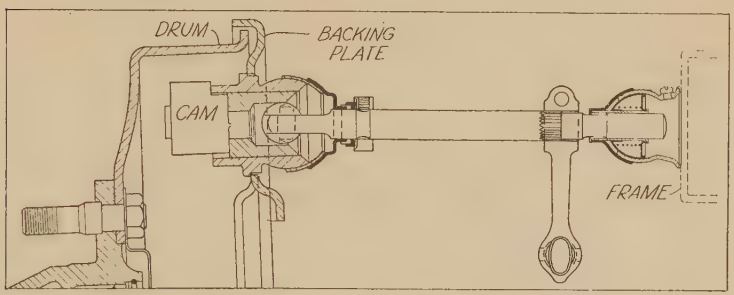


FIG. 35.—OPERATING MECHANISM OF PERROT FRONT-WHEEL BRAKE.

slowly, and that the difference between the speeds of the driving and driven shafts depends upon the angle between shafts.

Now suppose that the brake shaft of one of the front-wheel brakes and the cam connected to it are making a small angle with each other and that the position of the yokes of the universal is such that there will be an increase in the speed of the transmitted motion if the shaft is turned to apply the brake. With a small angle between shaft and cam axes the cam will advance angularly a certain small amount relative to the shaft while the latter is turned around its axis through a definite angle, while with a larger angle between axes the angular advance of the cam will be greater. But the total angular movement of the cam around its axis will be the same whether the cam is first deflected through a given angle and the shaft then turned around its axis through a

definite angle, or whether the shaft is first turned and the cam then deflected. Therefore, since the cam will advance more as the result of shaft rotation when it makes a larger angle with the shaft, it follows that increasing the angle between shaft and cam after the rotation of the shaft has been completed will further advance the cam around its axis.

In steering the inner wheel always deflects through a larger angle than the outer wheel, and the cam of the inner wheel therefore will be advanced more with relation to its driving shaft than the cam of the outer wheel. Since—owing to the absence of means of equalization—both shafts always

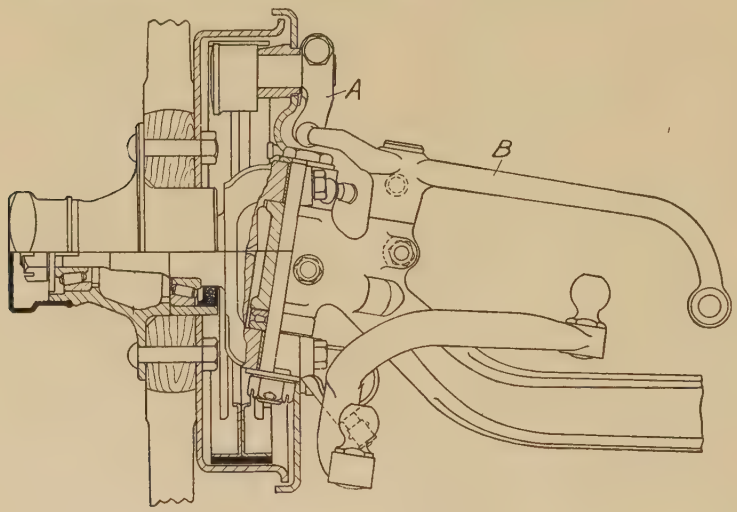


FIG. 36.—BENDIX LEVER MECHANISM.

occupy the same angular position, if the cam of the brake on the inner wheel applies the shoes firmly to the drum, the cam of the brake on the outer wheel, being less advanced, owing to the smaller deflection of this wheel, will not apply the brake so firmly, and this wheel therefore will be free.

The Perrot brake mechanism is somewhat expensive to manufacture, and its use in this country is confined to cars in the higher-priced classes. For lower-priced cars the manufacturers who control the Perrot U. S. patents, the Bendix Brake Corp., manufacture another type of front brake with a lever mechanism, as illustrated in Fig. 36. The brake also is cam-operated and the shaft of the cam just outside its bearing carries a short lever A which is engaged by the ball end

of a double-armed lever B fulcrumed on the front axle in such a manner that the center of the ball at the end of the horizontal lever lies in the axis of the knuckle pin when the brake is applied. Wear of the lining, of course, will cause the center of the ball for the "on" position to move away from the axis, and in order to make it possible to readily restore it to that position, a suitable adjusting means is provided by which the lever is moved angularly with respect to its shaft. As already mentioned, one type of adjusting mechanism comprises a worm and wheel.

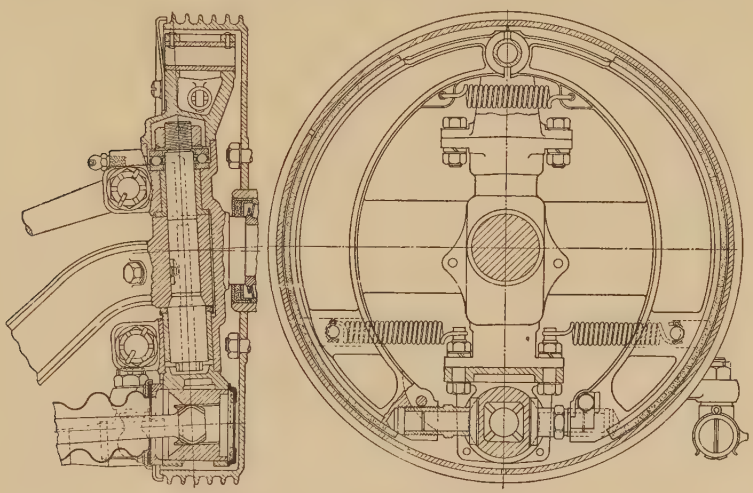


FIG. 37.—ISOTTA-TYPE OF FRONT BRAKE (Pierce Arrow).

Isotta Type of Front Brake—Another type of front-wheel brake, introduced by the Italian firm of Isotta-Fraschini about 1909, is illustrated in Fig. 37 as applied on the Pierce-Arrow car. The cam is located at the bottom of the brake and is supported by a bearing in a bracket bolted to the steering knuckle. The shaft operating this cam is supported in bearings on the front axle center; it has no rigid connection with the cam, but a sort of pivot connection which permits the cam to turn with the front wheel in steering. The brake rods in this case come inside the car frame.

A number of designs of front-wheel brake have been worked out in which the operating motion is transmitted to the shoes by means of rods passing axially through the knuckle pins. Pressure on these rods will apply the brakes, while springs between shoes or around the rod will release them.

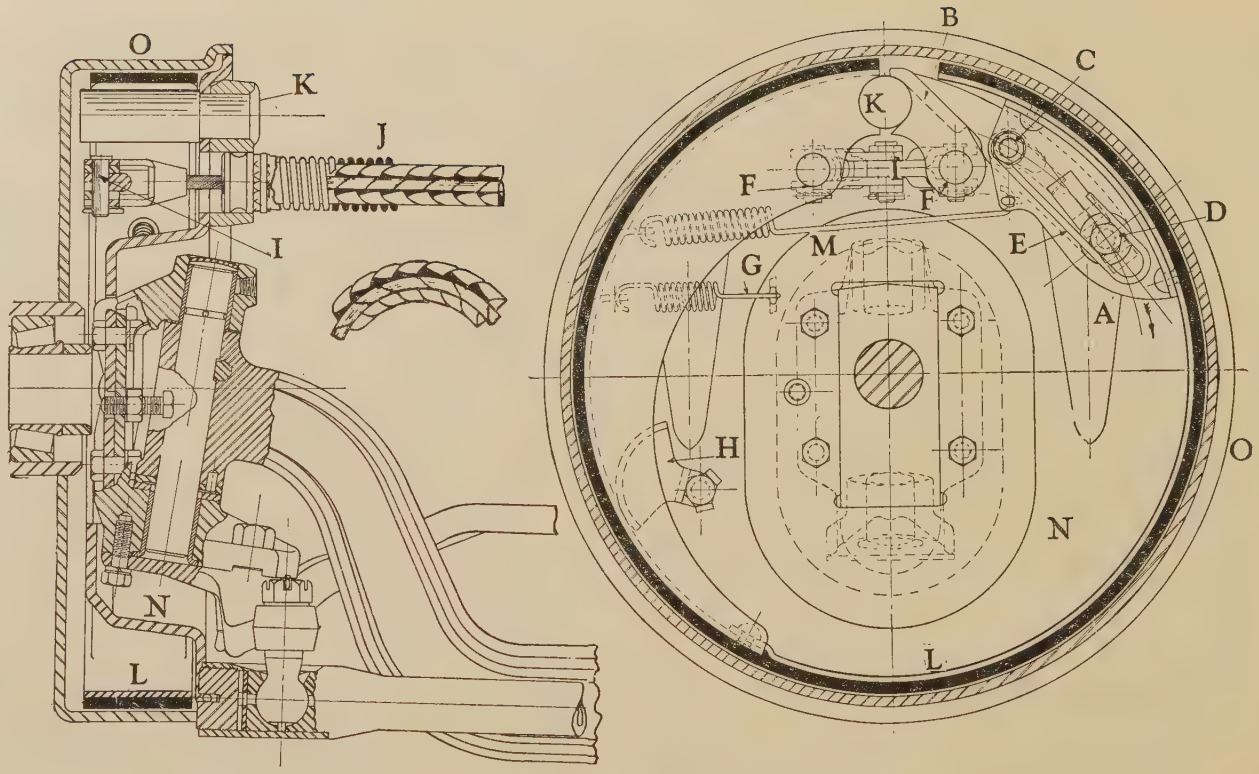


FIG. 38.—MIDLAND STEELDRAULIC BRAKE, WITH DETAIL OF FLEXIBLE CONDUIT.

The most prominent user of this type of mechanism in the United States is the Ford Motor Co.

Midland Steeldraulic Brake—A front-wheel internal brake of the self-energizing type is manufactured by the Midland Steel Corp. of Detroit under the name of Steeldraulic (Fig. 38). In this brake the applying force is transmitted through a flexible mechanism or conduit, not unlike the Bowden wire mechanism in principle. The expanding member, which extends over nearly the whole circumference of the drum, is made up of a rigid pressed-steel shoe of channel section, and a rolled-steel, flexible band, the two being united into a single member by riveting and welding. The shoe covers a slightly smaller portion of the circumference than the band.

For the expansion of the brake, use is made of a toggle mechanism, whose links move in a horizontal plane. The toggle links connect to two fulcrum pins, of which one is supported in the flanges near one end of the shoe, while the other is connected to the end of the band through the intermediary of an adjuster. To the band are riveted two angular brackets between which the adjuster moves. Adjustment is made by means of a wedge block, which is moved forward by a screw.

Application of the brake is effected by exerting a pull on the middle of the toggle through the "conduit." The "conduit" consists of a steel cable surrounded by a number of cupped screw-machine parts held together by spring pressure. Each unit can move relative to the units adjoining it, hence the assembly may bend as required when the car is being steered, yet the length along the center line will always remain substantially the same. The "conduit" is covered by a water-tight boot to retain the graphite grease with which it is packed to ease the movement of the cable.

The right end of the conduit assembly fits into a bracket fastened to the frame, while the other end is fastened to the anchor pin bracket on the backing plate. The right end of the cable terminates in a fitting which receives the brake rod, while the other end terminates in a yoke connecting to the toggle at its middle point. One feature of this brake is that, owing to the large arc of contact between brake and drum, the self-energizing effect is quite large. Besides, the toggle gives an increasing mechanical advantage or increasing ratio the more nearly its links come into line. It is also claimed that when describing curves greater resistance to the motion of the cable on the outer side, due to the much sharper bend of the conduit on that side, causes the outer brake to be applied

less firmly and prevents it from locking. Rear brakes are made of the same general type, and the system is applied in the form of four-wheel brakes only. No equalizers are used. Owing to the self-energizing effect, the clearance can be made comparatively large without the brake requiring excessive pedal pressure.

In Figure 38 A represents flanges secured to the end of the brake band; B is the brake adjuster; C, the adjuster clamp bolt; D, the adjuster screw; E, the adjuster safety clip; F, toggle fulcrum pins; G, spring between brake shoe and backing-up plate; H, brake adjuster stop; I, toggle; J, flexible conduit; K, anchor pin; L, brake shoe; M, shoe retractor spring; N, backing-up plate; O, brake drum.

Hydraulic Brakes—The complications involved in the mechanical transmission of motion to brakes on the front wheels led to the introduction of brakes in which the force of application is transmitted hydraulically. The best-known system of this kind is the Lockheed, which is being exploited by the Hydraulic Brake Co. of Detroit. Hydraulic brakes are used as four-wheel brakes only.

In the earlier applications the Lockheed hydraulic brakes were of the external band type for the most part, while more recently the internal shoe type has had the preference. The earlier systems consisted of a master cylinder containing a piston which could be forced into it by pressure on the brake pedal; a cylinder contracting the band for each brake; a supply tank with pump by means of which the supply of fluid in the system could be replenished after a certain amount had been lost through leakage, and a line connecting the master cylinder with all of the operating cylinders and with the supply tank.

The system being filled with the brake fluid (a 50 per cent solution of castor oil in alcohol, neutralized), when the operator presses on the pedal, he forces the piston into the master cylinder and thereby puts the fluid in the system under pressure. Fluid then flows through the line to all four brake-operating cylinders. These cylinders contain two pistons each and the fluid enters at the middle. Each of the metal pistons is provided with a rubber cup on the inside and there is a rubber boot between the piston and the end of the cylinder, these rubber members serving to minimize leakage of the fluid. Each of the pistons connects through a double-armed lever to one end of the brake band, and when fluid enters the operating cylinder and forces the pistons apart, the levers pull the ends of the band together and

contract the band on the drum. The band is released by a pair of coiled springs which force its ends apart.

The supply tank is usually mounted on the forward side of the dashboard, and a tube from it connects to the main line from the master cylinder at some convenient point. Centrally within the tank there is a rod with a finger wheel which serves both as pump plunger and valve. When it is desired to replenish the supply of liquid in the system, the rod is first turned left-handedly a number of times to unscrew it, whereby the needle valve at the bottom of the tank is

FIG. 39.—INTERNAL HYDRAULIC BRAKE (Lockheed).

opened. By then working it up and down, fluid is pumped into the system, and when enough has been forced in, the line is closed again by screwing the valve down to its seat.

In the case of internal shoe brakes the piston and the operating cylinder push directly against the "toe" ends of the shoes, as shown in Fig. 39. In this particular case the shoes are supported on independent anchor pins and are withdrawn by a spring near their toe end. Near the toe end of each shoe there is an adjustable stop of spiral form which serves to equalize the release motions.

Figure 40 shows a compensating type of master cylinder which obviates the need for replenishing the system by means of a pump at intervals. The master cylinder is bolted into the lower part of the supply tank, and the brake pedal is supported on this tank, which in turn is mounted on the chassis frame. It will be seen that the piston is provided with a rubber cup which is pressed against it by a coiled spring through the intermediary of a pressed-steel cup.

At the end of the master cylinder where the fluid passes into the line when the brake is applied, there is a double valve. One of these valves, which is held to its seat by the

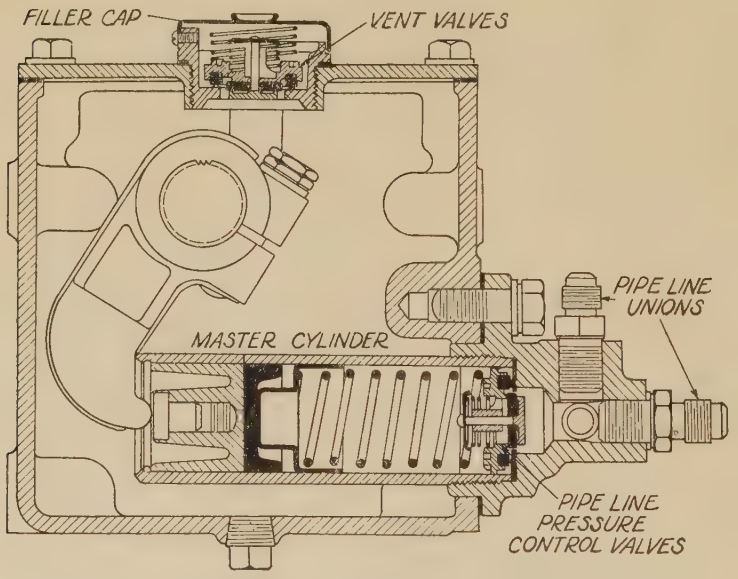


Fig. 40.—Automatically Compensating Master Cylinder Submerged in Supply Tank.

large coiled spring within the cyinder, opens when an increase in temperature expands the fluid in the line, and allows the excess fluid to pass through the cylinder into the supply tank, through a small port in the cylinder wall which is uncovered by the rubber cup when the piston is in the "off" position. If the fluid in the line contracts again due to a decrease in temperature (or if there should be leakage at the operating cylinders), fluid from the working cylinder (which is at all times kept filled from the supply tank by gravity) will enter

the line through the second of the two valves, which opens toward the line and is held to its seat by a very light spring under compression.

The two valves mentioned do not take part in the ordinary operation of the brake, but serve merely to keep the line filled at all times. The line is kept under a constant pressure of 8 lbs. per sq. in. by the large spring in the master cylinder, which has the advantage that it keeps the rubber cups in the operating cylinders expanded, thus preventing leakage. Since the master cylinder is completely submerged in fluid, there is no possibility of air getting into the system. In some installations the master cylinder is separate from the supply tank, which, however, does not affect the self-compensating feature.

A feature of hydraulic brakes is that complete equalization is effected at all times without any special mechanism.

One of the difficult problems in connection with hydraulic brakes was found to be the development of a flexible hose for connection to both front and rear brakes, that would not expand materially under the pressures employed in the system. A hose used for this purpose consists of an inner rubber core, surrounded by four layers of fabric with an outer rubber covering, the whole being vulcanized. Into this flexible hose is inserted a close-wound steel spring of a slightly larger outside diameter than the inside diameter of the rubber hose. This steel spring holds the wall of the hose under tension, and prevents its further expansion.

Bendix-Perrot Three-Shoe Brake—In order to take advantage of the self-energizing principle in a shoe brake, Henri Perrot in France, in collaboration with the Farman Company, developed a two-shoe internal brake in which the shoes were articulated together, the remaining end of one shoe being connected to an anchoring pin, and that of the other, acted on by the cam. Such a brake, while very powerful in checking forward motion, has practically no power to stop reverse motion. The Bendix Corporation in this country, in further developing the Perrot brake inventions, therefore adopted the three-shoe construction illustrated in Fig. 41. Here two shoes are articulated together, while the third shoe is independent. During forward motion, the friction on the shoe which is not directly connected to the anchoring pin, tends to apply the shoe to which it is articulated, harder to the drum. During reverse motion the effect, of course, is the opposite, but the third shoe is quite powerful in checking reverse motion, the friction adding to the force applied through the cam in applying it to the drum.

Mechanical Boosters—Mechanical servo brakes or brake boosters are used to quite an extent in Europe. Their principle may be briefly explained as follows: Imagine a disc securely fastened to a shaft which rotates as long as the car is in motion. Adjacent to this disc there is another one which can be brought into frictional contact with it by pressure on a pedal. This latter disc is so mounted that it can rock around its axis. Connected to this disc is a lever arm from which connection is made to the brakes through the conventional linkage. The pull on the brake linkage will then be in direct proportion to the pressure of the pedal, and

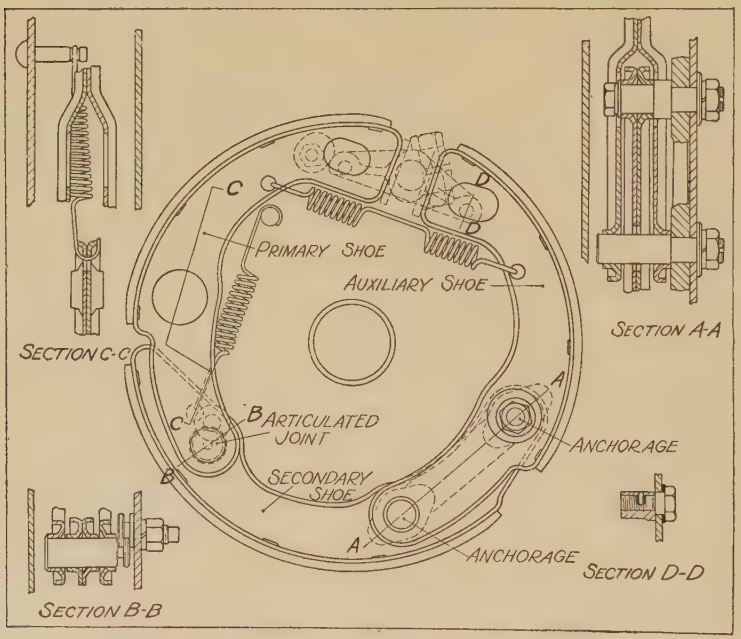


Fig. 41.—Bendix-Perrot Three-Shoe Brake.

owing to the fact that the axial motion of application and release of the loose disc need only be very small, the mechanism can be readily so proportioned that the pull on the brake linkage will be several times as large as if it were acted on by the pedal directly. The force of application then comes largely from the momentum of the car.

Such a booster when applied to the propeller shaft is quite effective for forward motion, but absolutely useless for re-

verse motion. It is possible to make mechanical boosters that are equally effective for both directions of motion, but these naturally are rather complicated. Such double-acting boosters are used on the Rolls-Royce, the Renault and several other Continental cars.

Vacuum Boosters—As long as the engine is running, there is a vacuum in its inlet manifold, which can be used to help apply the brakes. When the engine is running slowly under full throttle the vacuum is only small, but when it is desired to stop, the throttle is naturally closed, and the vacuum then rises to about 10 lbs. per sq. in. A vacuum servo consists of a cylinder with a piston in it. One end of the cylinder can be placed in communication with the inlet manifold through a connecting tube with a suitable valve in it, while the other end is open to the atmosphere. The atmosphere then pushes against the piston on one side with its full pressure of 14.7 lbs. per sq. in., while against the other side of the piston there is the reduced pressure in the inlet manifold, which may be as low as 5 lbs. per sq. in. abs. Thus the piston is forced toward the end connected to the inlet manifold with a pressure of up to nearly 10 lbs. per sq. in., whereby the brakes are applied.

It is, however, not sufficient to merely provide a valve to open one end of the cylinder to the inlet manifold, as that would give brake application without control. Some means must be provided whereby the difference in pressures on opposite sides of the booster piston is kept proportional to the pressure exerted on the pedal, up to the full difference between atmospheric and manifold pressure, and it is the means by which this is accomplished that distinguish the different vacuum servos.

Figure 42 illustrates the Westinghouse (Dewandre) vacuum servo. Within the vacuum cylinder of 4½ in. diameter there is a piston which is forced outward by a spring. The piston connects by a chain to a floating lever L. This is a double-armed lever which is fulcrumed on the end of the single-armed lever M. A pull rod N from the brake pedal connects to lever L, and a rod O from lever M connects to the brakes. At its upper end lever L is provided with a large eye which surrounds the fulcrum pin of lever M with a certain amount of clearance. From the upper end of lever L extends a rod P to the double-armed lever Q which operates the double valve R. When this valve is moved to the right it places the vacuum cylinder in communication with the inlet manifold, while when moved to the left, it opens the cylinder to the atmosphere. Owing to the clearance of the eye of lever L,

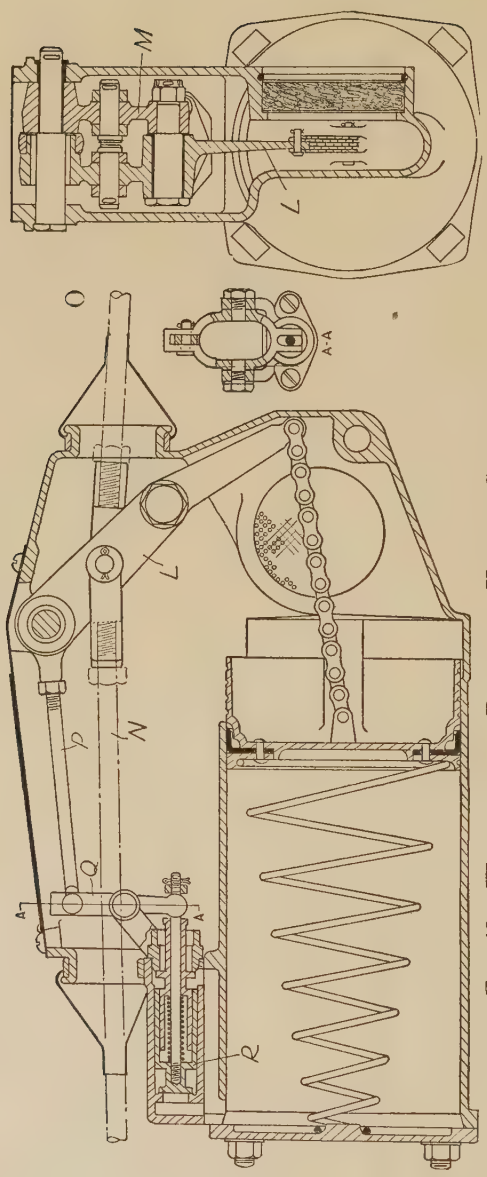


FIG. 42.—WESTINGHOUSE-DEWANDRE VACUUM SERVO SHOWN IN SECTION.

this lever has a limited motion which does not affect the valve.

Ordinarily when the brake is on "light," there is a certain vacuum in the brake cylinder, less than in the manifold, and the vacuum valve is closed. The fulcrum pin of lever M then is clear of the eye of lever L, or at least does not press hard against that lever. Considering the lengths of the different lever arms, if we regard the pull on the pedal rod N as equal to unity, then the pull on the chain is 0.57, and the force on the fulcrum of lever L or the pull of this lever on lever M is 1.57. Since the brake rod O connects to M midway between its fulcrum and the fulcrum of lever L, the pull on rod O is $2 \times 1.57 = 3.14$. This pull is directly in line with that on N, and the motions of the two rods also are equal. Therefore, in this particular case the pressure on the pedal is multiplied in the ratio of 1 to 3.14 and it can be made either larger or smaller by simply changing the proportions of the levers.

If the driver now presses harder on the pedal, the balance of the system of levers will be temporarily disturbed. The upper end of lever L will move to the left, which will result in opening the vacuum valve, whereupon more air is sucked from the brake cylinder. This causes a harder pull on the chain, whereby the equilibrium is restored again, and the vacuum valve closes again. If the driver increases the pressure on the pedal beyond a certain point, the vacuum valve is permanently held open, and the full vacuum of the inlet manifold is active in the cylinder. Any further increase in the pedal pressure will add only that additional pressure to the force of brake application. When pressure on the pedal is released, the retracting spring draws the pedal back and opens the atmospheric valve, thus allowing air to enter the brake cylinder.

Should the engine be stalled when the brake is applied, the pressure on the pedal is transmitted directly, the same as if there were no power brake.

Hydraulic Servos—Hydraulic mechanisms for operating brakes by the kinetic energy of the moving vehicle have been used by a number of firms, and a servo of this type in which the force of brake application is proportional to the pressure applied to the pedal (beyond a small fixed pressure required to lift the valve of the servo from its seat), has been developed by E. G. Staude of Minneapolis. It consists of a gear-type oil pump, a pressure cylinder and valve, all parts being arranged in a single unit which is mounted on the side of the transmission. The pump moves oil as long as the car

is in motion, the oil ordinarily flowing from the pump outlet directly back to the inlet. When the pedal is depressed, this passage is restricted, and a pressure is set up in the system. The oil then presses against the piston, whose rod bears against an arm on the pedal, and the piston pressure is thus added to the pressure of the driver's foot on the pedal.

Air Brakes—Air brakes, which are generally credited with having made high-speed railroading possible, are being used also on motor trucks and buses, and particularly on rail cars. A small air compressor is fitted, being driven from the engine, and keeps an air tank constantly under pressure. By means of a special valve the air can be admitted to diaphragm chambers containing diaphragms of rubberized

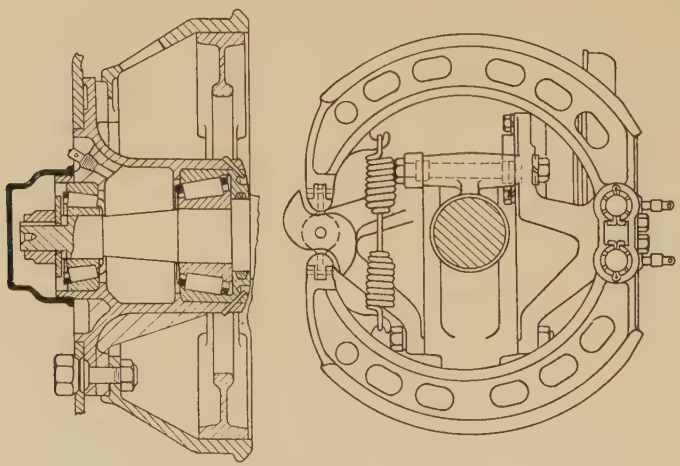


FIG. 43.—METAL-TO-METAL BRAKE FOR AIR-OPERATION.

fabric. When compressed air is admitted to the chamber on one side of the diaphragm, the latter presses against a metal plate on the other side of it, this plate being secured to a rod which is connected to the brake rigging. Either a hand valve or a foot valve can be used, hand valves having been found most practical on trucks and fire apparatus, and foot valves on buses. All valves have the characteristic that they admit pressure to the diaphragm chamber in direct proportion to their displacement from the "off" point. One reason for using diaphragm chambers instead of cylinders is that this prevents all leakage in the operating members.

Usually the diaphragm chamber is mounted directly on

the axle housing or steering knuckle, close to the brake, and the diaphragm plate connects through a short rod to the brake lever. Metal-to-metal brakes have been found best adapted for air operation. The coefficient of friction between metals (cast iron and steel) is considerably smaller than that between metal and the regular fibrous brake linings, but this is compensated for by the use of greater pressures. Internal shoe brakes are used, and a worm-type of adjustment is provided between the camshaft and lever to compensate for wear.

Air brakes have proven quite popular on high-speed buses, for one reason because the compressed air can be used to advantage also for other purposes, such as operating door engines.

Air brakes also lend themselves well to use on trailers, since it is only necessary to make an air-hose connection between truck and trailer when the two are coupled up, to make the trailer brake effective. A pressure tank is carried on the trailer, and the design of the installation on the trailer is such that if the latter becomes uncoupled for any reason, its brakes are set automatically, thus eliminating the possibility of its running down hill. A metal-to-metal brake for operation by air diaphragm is shown in Fig. 43. This brake is cam-expanded, the cam being of the "uniform lift" type.

CHAPTER XII

Front Axles

Form of Front Axle—The front axles of practically all modern motor vehicles are of the divided type, comprising an axle center and two steering knuckles, the latter being pivoted to the former. In most of the earlier vehicles the front axles were absolutely straight, their centers being in line with the spindles of their steering knuckles. Later, when the engine was placed in front, and either it or a drop-type of cross-member of the frame—carrying the radiator—came directly over the axle, it became necessary to provide the latter with a "center drop," to ensure clearance between axle and frame or engine under conditions of maximum spring deflection. Still later, when the frame and the center of gravity of passenger cars had to be lowered to ensure greater stability and safety at high speeds, the center portion of the axle center, including the two spring seats, was dropped. These three stages in axle development are illustrated in Fig. 1.

Bending Moments on Axle—To obtain the required strength with a minimum weight of material, it is necessary to make a careful study of the loads and moments on the various parts of the axle under all operating conditions, and to design the parts in accordance with the results thus obtained.

In a vehicle or tractor in which both driving and braking are effected through the rear wheels, the front axle is subjected to rather simple stresses. While the vehicle is at a standstill the axle serves merely as a carrying member, and is subjected only to bending stresses in a vertical plane. The distribution of these stresses is much the same as in a rear axle housing. There are two loads causing bending stresses, viz., the weight of the axle itself, and that part of the spring-supported weight of the chassis and of the load which is carried on the front axle. As compared with the latter, the

weight of the axle is small (3 to 4 per cent) and may safely be neglected. The weight supported on the axle by the springs produces bending moments which may be represented by a quadrilateral whose base line represents the wheel tread, the top line the distance between spring centers, and the height, the bending moment at the particular point along the axle. Thus the vertical bending moment increases from nothing in the vertical plane through the center of the wheel track, to a maximum at the center of the spring seat, remains constant between spring seats, and then decreases uniformly to nothing in the vertical plane through the center of the other wheel track.

When the car is in motion the resistance encountered by the front wheel creates a horizontal bending moment on the front axle. Ordinarily this is quite small as compared with

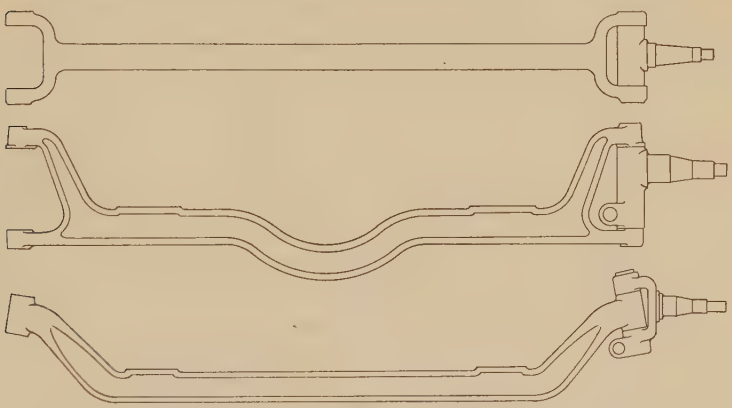


Fig. 1.—Three Designs of Front Axle.

the vertical bending moment, the ratio between the two being equal to the traction coefficient (about 2 per cent). Under difficult traction conditions, and on up grades, this horizontal moment increases, and it reaches its maximum value when the wheel strikes an obstruction, but this maximum is incapable of calculation.

The distribution of horizontal stresses along the length of the axle depends upon the connection between the axle and the chassis frame. If this is by means of semi-elliptic springs which are rigidly clipped to the axle, the horizontal bending moment starts from zero at the knuckle pivot axis and increases uniformly to a maximum at the spring seat. There

is no horizontal bending moment of any consequence between spring seats. If, on the other hand, the frame is supported on a transverse spring and the drive is transmitted to the axle through radius rods which have universal (ball and socket) connections with it, then the horizontal moment due to resistance encountered by one wheel reaches a maximum value at the point of attachment of the radius rod to the axle and will remain constant between points of attachment of the two radius rods if the resistances encountered by the two wheels are alike.

Torsional Moments on Axle Center—The above discussion covers the case of a straight axle or any design in which the portion of the axle center outside the spring seats is substantially in line with the wheel spindles. When there is a pronounced drop from the wheel spindles to the spring seats, the axle center forms a sort of crank, and that section of it directly outside the spring seat is subjected to a torsional moment due to the resistance encountered by the front wheel. This moment is equal to the product of the resistance into the length of the crank arm (*Fr.* Fig. 2).

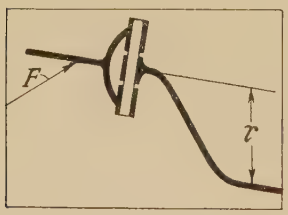


FIG. 2.—DIAGRAM SHOWING TORSIONAL MOMENT ON DROP AXLE DUE TO WHEEL RESISTANCE.

When the front wheels are driving wheels, or when brakes are applied to them, there is another torsional moment to be considered, in addition to that already discussed. The limiting torsional moments in driving and braking are of the same order of magnitude, and since the use of front-wheel brakes is much more common than front wheel drive, we will here consider the torsional moments due to braking.

Front-Wheel Braking Torque—When the front wheel is practically locked by the brake, the resistance to its motion under optimum braking conditions (dry, hard road surface) is about $0.6W$, where W is the weight on the wheel. The torsional moment M at the section of the axle center (Fig. 3) close to the steering head is then $0.6WR$, where R is the wheel radius. This is much greater than the bending moment Wl_1 at this section. In a modern passenger car, R is about 15 in. and l_1 3½ in., which makes the maximum torsional moment almost three times the vertical bending moment and indicates that the section near the steering head should be made such that its resistance to torsion will be relatively

great. The best section from this point of view is a circular one.

At a section just outside the spring seat, the torsional moment is less, viz., $0.6W\ (R - r)$, where r is the drop from the spindle axis to the center of that section. The bending moment, on the other hand, is much greater than at the steering head, and it predominates at this point. In an average passenger car r will be 5 in. which makes $R - r = 10$ in. Hence, the bending moment at the spring seat is $12W$ and the torsional moment $0.6W \times 10 = 6W$. Therefore, a section should be chosen which offers great resistance to bending.

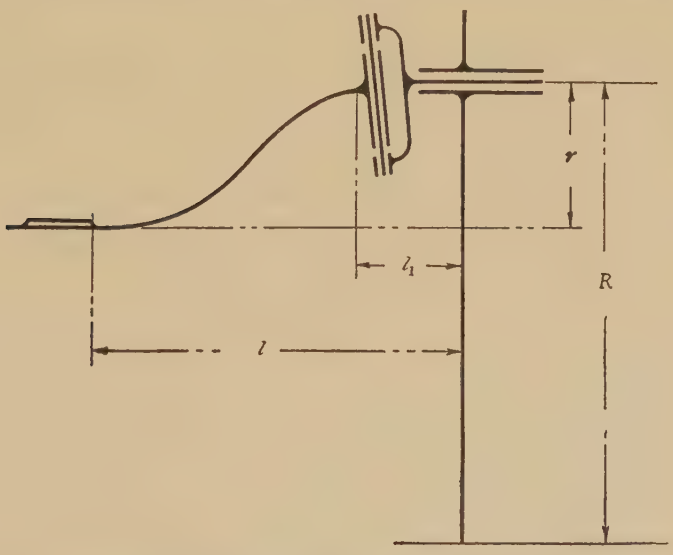


FIG. 3.—BENDING AND TORSIONAL MOMENTS ON SECTIONS OF AXLE-CENTER OUTSIDE OF SPRING SEAT.

One solution is to use a rounded rectangular or elliptic section at the spring seat and gradually change to a circle on approaching the steering head. This plan is quite popular in Europe. Another scheme consists in making the section rectangular, with gradually decreasing height as the steering head is approached, adding a flange at the top, which increases in width toward the spring seat, and really merges into the latter. This is to take care of excessive horizontal bending moments produced when the wheel strikes a large obstruction. It gives a T-section for the greater part of the

axle center outside the spring seat. Still another scheme consists in gradually changing from an I-section at the spring seat to a solid round section at the steering head.

An I-section, which is very advantageous where only bending stresses in one plane have to be withstood, is quite unsuitable for use where the stresses are chiefly torsional. In fact, the strength of an I-section under torsion is not materially greater than that of a rectangular section of the same height as the I-section and of a width equal to the thickness of its web, which makes about one-half of the material of the I-section entirely useless; and even the rectangular section of a height large compared with its width is very inefficient under torsion.

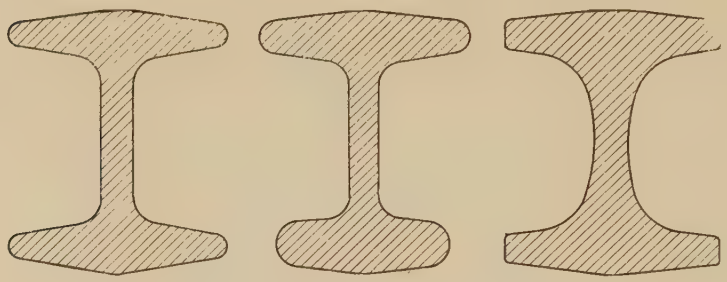


Fig. 4.—Three Sections Used for I-Beam Axles.

Center Sections—There are two principal types of front axle, the I-section axle and the tubular axle. The tubular front axle is an excellent design, especially where front-wheel brakes are to be carried, as it usually weighs less than an I-section axle for a given strength, but it is rather expensive to manufacture. As a rule, chrome-molybdenum steel tubing is employed, of from 1/8 to 5/32-in. wall thickness, and steering heads and spring seats are riveted on. In the heat-treated state this steel has an elastic limit of 180,000 lbs. per sq. in., and a maximum stress of 30,000 lbs. per sq. in. under static loads can be allowed with it.

I-section axle centers are drop-forged from medium carbon steel (S. A. E. No. 1035 or No. 1040). These steels, when suitably heat-treated, have an elastic limit of 60,000 to 75,000 lbs. per sq. in., and a tensile strength of 90,000 to 105,000. In designing axle centers of this material, it is customary to allow a stress under full static load of 10,000 lbs. per sq. in. in the case of trucks fitted with solid rubber tires; about 12,000 lbs. in the case of high-speed vehicles on pneumatic

tires, and up to 15,000 lbs. in the case of moderate speed vehicles on pneumatic tires, such as city buses.

I-Section Axles—The form of I-section employed varies somewhat in different designs. Figure 4 shows three different sections, all reduced to the same height. The flanges must be given about 7° draft, so the forging can be withdrawn from the dies. It is interesting to mention in this connection that whereas previously all I-section front axles were completely drop-forged, many are now produced partly by a rolling process. The process starts with a billet of square section, the center portion of which is rolled into an I-section. Then this part is "broken down" to obtain the necessary center drop, and finally the ends are upset and forged to shape in dies, one at a time. Rolling subjects the material to mechanical working and increases its strength.

In calculating the strength of I-sections the author makes use of an equivalent section with rectangular web and flanges, as shown in Fig. 5, which gives fairly accurate results. If a higher degree of accuracy is desired, use may be made of any of the methods for finding the moments of inertia of irregular sections, explained in textbooks on theoretical mechanics.

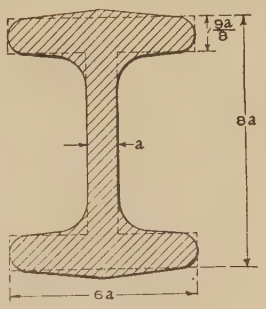


FIG. 5.—SIMPLE GEOMETRICAL FIGURE SUPERPOSED ON EQUIVALENT AXLE SECTION.

The section shown in Fig. 5 is a good average. Denoting the thickness of the web by a, the width of the section is $6a$ and the height slightly over $8a$. The ends of the flanges are semi-circular, of radius $a/2$, and the fillet between web and flange has a radius a. The sides of the flanges are inclined 7° to give the necessary draught. The dotted figure represents an equivalent geometric section, and in this the height is exactly 8 times the thickness of the web. The thickness of the flange of the geometric equivalent section is $9/8a$. The moment of inertia of such a section is

$$\frac{6a \times (8a)^3 - 2 \times 2.5a \times (5\tfrac{3}{4}a)^3}{12} = 188a^4.$$

The distance c of the outermost fiber from the neutral section being $4a$, the section modulus is

$$\frac{177a^4}{4a} = 44.25a^3.$$

The moment of inertia of this same section around a vertical axis is

$$\frac{2 \times \frac{3}{8}a \times (6a)^3 + 5\frac{3}{4}a \times a^3}{12} = 49.31a^4$$

and since the distance c in this case is $3a$, the horizontal section modulus is

$$\frac{49.31a^4}{3a} = 16.44a^3.$$

The section modulus is a measure of the strength of the section, and the section shown in Fig. 5 therefore is three times as strong vertically as horizontally.

The end portion of the axle center may be regarded as a cantilever, and the equation of its external and internal moments is

$$Wl = SZ,$$

where W is the reaction of the ground on the wheel (equal to the load on the wheel); l, the distance from the vertical plane through the center of the wheel track to the center of the adjacent spring seat; S, the maximum stress in the material of the section between spring seats, and Z, the section modulus of the axle section around a horizontal axis. To give a rough idea of the capacities of axles of various sizes, the following table has been compiled, giving the vertical bending moments which axles with sections of the form shown in Fig. 5 will sustain when stressed to 12,000 lbs. per sq. in.

Height of section (ins.).	1½	1¾	2	2¼	2½	3	3½
Allowable moment (lb.-in.)	4660	7400	11,030	15,720	21,600	37,300	59,200

The allowable moment divided by the distance l gives the maximum load which the wheel may carry.

Section Near Steering Head—As already pointed out, the stress on the axle center near the steering head is mainly torsional, and the maximum torsion occurs when the front wheel is locked (or nearly locked) by the brakes. Assuming the maximum road adherence to be 0.6, the tangential force on the wheel rim is $0.6W$, and the torsional moment on the section then is $0.6WR$, where R is the wheel radius. Making use of the equation for the torsional strength of round sections, we have

$$0.6WR = 0.196d^3S,$$

where d is the diameter of the section; and S, the allowable stress. In this case we can use a higher value for S than when calculating sections subjected chiefly to bending loads, because the torsional stress occurring when the wheel slips on dry, hard surface is practically the limit, whereas the bending stress is greatly increased by road shocks. Rear axle members subjected to similar stresses are calculated with a factor of safety of two, based on the elastic limit in torsion. It is advisable to make the factor of safety for the front axle somewhat greater, because of the greater personal risks involved in front axle failure. A stress of 20,000 lbs. per sq. in. may safely be figured with for the section of the steering head if the axle is forged of No. 1035 or No. 1040 steel and suitably heat treated. The diameter of the round section required then is

$$d = \sqrt[3]{\frac{3WR}{S}} = \sqrt[3]{\frac{WR}{6{,}667}}.$$

It is well to also check the stress induced in this section by the maximum bending moment. The load on the front axle, of course, is not known beforehand, but it can usually be sufficiently closely estimated from the load on front axles of similar existing vehicles.

Spring Pads—Drop forged axle centers are always forged with integral spring pads. Each pad has four holes drilled through it for the spring clips. The distance apart of the centers of the holes in the direction of the axle length is equal to the width of the spring plus the diameter of the hole, and the distance apart in the other direction is made about the same. After the clip-hole centers have been determined, arcs are struck from these centers to radii about 2.5 times the clip-hole radius, which form the "corners" of the pad. The front and rear edges of the pad are made straight, while the side edges are given draft. The top of the pad is milled off to a level of 1/16–1/8 in. above the highest point of the section adjacent to the pad, and a suitable finish allowance must be made. At its front and rear edges the pad should have a depth slightly greater than the average thickness of the top flange, and rather more than the usual draft is allowed at the bottom of the pad, so that its depth where it joins the web of the section is about twice as great as at the front and rear edges. In addition to the four holes for the spring clips a central hole is always drilled in the pad, to accommodate the head of the spring center bolt. Where the front axle takes braking torque the pad is sometimes

braced by vertical ribs joining it to the lower flange. Figure 6 shows a typical spring pad.

Steering Heads—There are a number of different designs of steering head. Previous to the advent of front-wheel brakes, the most popular type was the Elliott. With this, the ends of the axle center are forked, while the steering knuckle usually is T-shaped. A steering head and knuckle of the Elliott type, combined with a tubular axle center, are shown in Fig. 7. The drop-forged steering head is secured to the tubular axle center by eight rivets. The top of the wheel spindle is practically on a level with the top of the vertical part of the knuckle, which seems to be necessary in this case in order to afford room for the brake cylinder and mechanism above the steering head. The knuckle is forged with a flange to which the brake backing plate is secured, and it will be noticed that this plate is made irregu-

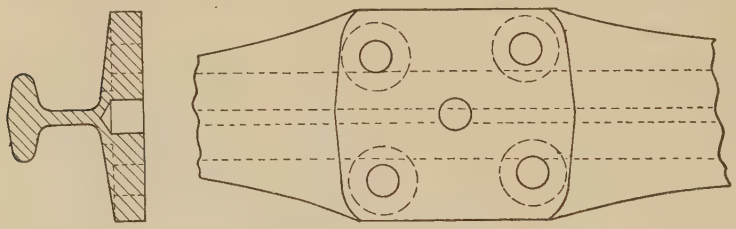


Fig. 6.—Spring Pad.

lar in section, the reason being that it must clear the steering head. Cross-pins prevent the knuckle pin from turning in its seat. The pin has bushed bearings in both arms of the steering head. Thrust load on the knuckle is taken on a ball thrust bearing.

Reverse Elliott Head—A second type of steering head is known as the reverse Elliott, the end of the axle center forming a vertical or nearly vertical cylindrical boss and the knuckle being forked. This type has come into the lead since the adoption of front-wheel brakes, as it makes it possible to rivet a flat backing plate for the brakes directly to a flange machined on the knuckle.

A reverse Elliott-type of axle end and steering knuckle is shown in Fig. 8. The knuckle pin, which is tubular, is held in position in the steering head by a cross-pin or bolt passing slightly below its surface. It will be seen that the axle center changes gradually from an I-section to a round solid section at the end. In this case the thrust is taken on

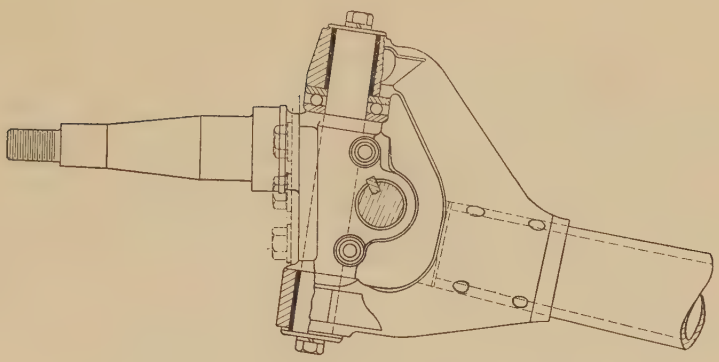


FIG. 7.—ELLIOTT-TYPE STEERING HEAD.

plain thrust washers which are protected by a dust cap. The bearings for the knuckle pin in both prongs of the knuckle are bushed. Lubrication of the various bearings is provided for by an oil-gun fitting in the upper prong of the knuckle. The lower prong is forged with a large boss into which the knuckle arm is fitted by bolting and keying.

Lemoine Axle End—What is sometimes referred to as the Lemoine-type of axle end is illustrated in Fig. 9. The knuckle is of very simple form and lends itself to a secure fixing of the arm without the use of a boss projecting beyond its natural outline. This type is now seldom met with, because it tends to raise the axle, whereas modern conditions require it to be lowered.

A design that is essentially the reverse of the Lemoine

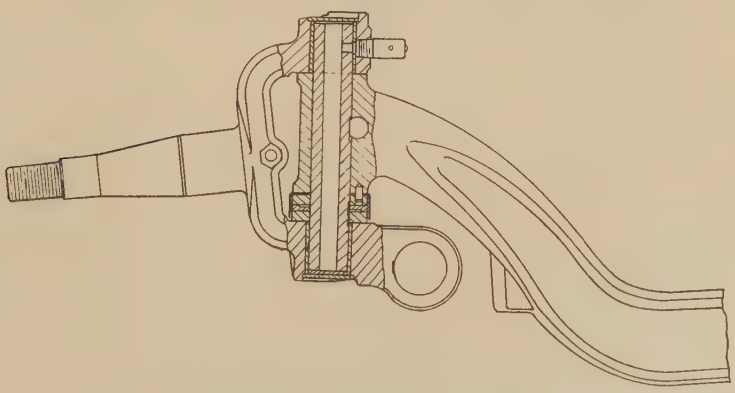


FIG. 8.—REVERSED ELLIOTT-TYPE STEERING HEAD.

has been used by Marmon and is illustrated in Fig. 10. The steering knuckle is of L-shape, as in the Lemoine, but the knuckle pin portion extends downward rather than upward from the wheel spindle portion. This gives a natural drop to the axle, and therefore overcomes the objection raised to the Lemoine type. The axle in this case is hung from the steering knuckle, its weight being supported on an anti-friction bearing. Since the steering head is oil-tight at the bottom, the knuckle bearings can be very effectively lubricated.

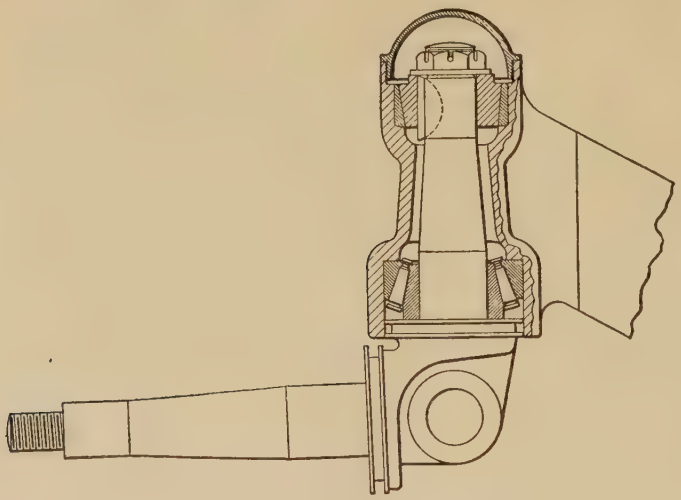


FIG. 9.—LEMOINE-TYPE STEERING HEAD.

Steering-Knuckle Materials — Steering knuckles and knuckle arms are made of alloy steel, as a rule. There are good reasons for choosing materials of higher grade for these parts. In the first place, failure of a steering knuckle or knuckle arm while the car is traveling at considerable speed is a very serious matter; and, secondly, owing to the rather complicated forms of these parts there is greater danger of localization of stress than in the axle center. A low nickel-chromium steel, No. 3135, is much used, but a 3½ per cent nickel steel, No. 2330, and a chromium steel, No. 5130, also find application. Both knuckles and knuckle arms are generally made of the same material, and this is logical, since both are subjected to the same kind of stress.

Steering Knuckle in Hub Form—In most cases the steering knuckle comprises a (nearly) horizontal spindle on which

the wheel hub revolves on anti-friction bearings, but the reverse arrangement is also encountered, in which the horizontal portion of the knuckle is developed in the form of a hub containing bearings in which revolves a spindle having the wheel center formed integral or bolted to it. The usual object of this arrangement is to bring the knuckle pin axis into the center plane of the wheel, which tends to reduce the transmission of road shocks to the steering wheel and the strains on the steering mechanism. One such design, used on Mannesmann-Mulag trucks, is shown in Fig. 11. The

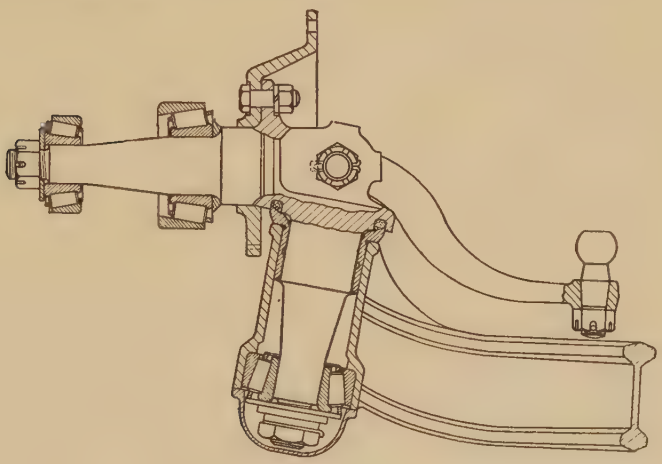


FIG. 10.—MARMON STEERING HEAD.

road wheel is of disc type, the disc being bolted to a center forged integral with a spindle. This latter is carried in roller bearings in a bearing shell provided with trunnions. In order to make it possible to get this bearing shell into position in the axle end, the lower arm of the latter is made with a split bearing. Thrust load is taken on hardened steel buttons set into the upper trunnion and an adjustable bushing in the upper arm of the axle end, respectively. This bushing carries an upward extension with a steep, square thread on it, a nut screwing over this threaded plug serving to apply the front-wheel brake. The knuckle arm is secured to the projecting portion of the lower trunnion.

Loads on Knuckle-Pin Bearings—The vertical thrust load on the knuckle is equal to the weight supported by the adjacent wheel. On vehicles fitted with balloon tires this load

is generally taken on a thrust bearing of the anti-friction type, in order to reduce the effort required in steering the car. The load is a steady one, and in selecting the size of bearing, one with a rated load capacity at least 50 per cent greater than the static thrust load should be chosen.

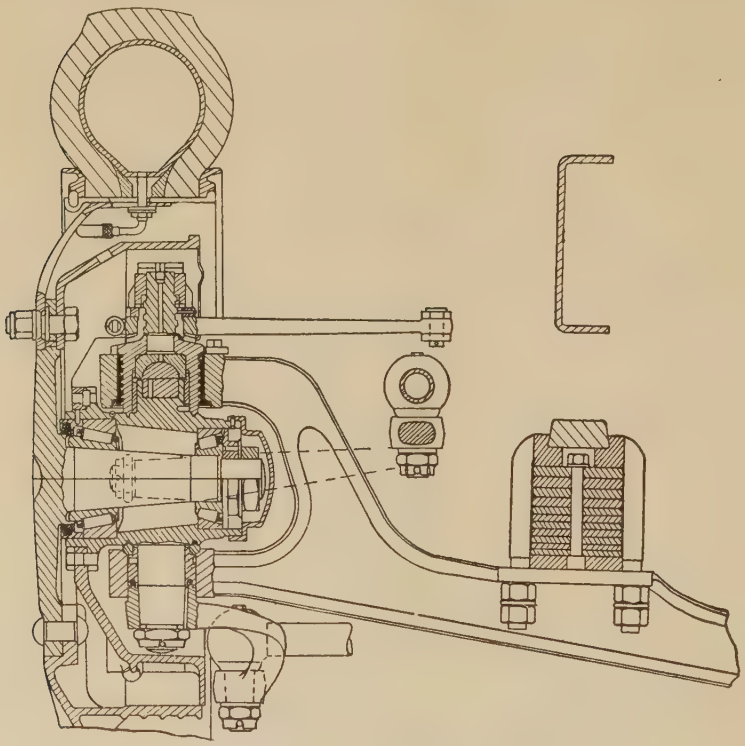


Fig. 11.—Hub-Type of Steering Knuckle.

The load on the radial bearings of the steering pivot may be calculated as follows (Fig. 12): Let P represent the maximum static reaction of the wheel on the knuckle spindle; a the distance from the center plane of the wheel to the axis of the pivot; l the distance between centers of radial bearings, and P' the load on each radial bearing. Then

$$Pa = P'l$$

and

$$P' = P\frac{a}{l}.$$

There are other loads on the radial bearings, such as those due to the traction resistance and to road shocks, but they are all proportional to that due to the static load, and may therefore be taken care of in the factor of safety, or in the unit bearing load allowed. The radial bearings can be so proportioned that the unit bearing load on them, calculated as explained in the foregoing, is from 400 to 500 lbs. per sq. in.

Inclined Steering Pivots—In all cars having front-wheel brakes the point of intersection of the pivot axis with the ground should be very close to the center of tire-ground contact. In most of these cars—and in some without front brakes—the pivot axis is inclined as shown in Figs. 7, 8 and 10. The distance between the point of intersection of the pivot axis with the road and the center of tire-ground contact forms the lever arm at the end of which most shocks sustained by the road wheel act. The shorter this lever arm the smaller will be the strains imposed by road shocks upon the steering gear and its linkage, and the less the fatigue of the driver's arms. Although the term "irreversible" steering gear is widely used, few if any steering gears on motor cars are actually irreversible, and some of the shock is always transmitted to the steering wheel. An objection to strongly inclined steering pivots is that in any but the straight-ahead position the front wheels are considerably inclined, in which position they are not as resistant to vertical loads as in the vertical one. Moreover, when the wheels are returned to the vertical position the whole front part of the car must be raised, and this increases the resistance to the steering motion.

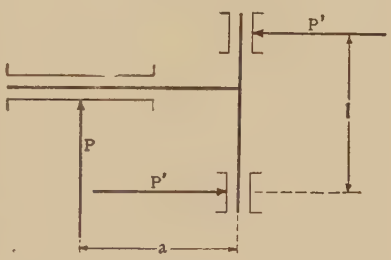


FIG. 12.—DIAGRAM OF KNUCKLE-PIN BEARING LOADS.

It has been found desirable, particularly with balloon tires, to have the intersection of the pivot axis with the ground come a short distance inside the center line of the track, rather than to intersect it, as that makes it easier to maneuver the front wheels when the car is standing still. If the pivot axis passes through the center point of ground contact, the motion between tire and road under the above condition is entirely a sliding motion, and as with balloon

tires the area of ground contact is comparatively large, the resistance to such motion is great. If the pivot axis strikes the ground some distance inside the center of the track, there will be partial rolling motion when the wheels are maneuvered with the car standing still.

Camber—Axle ends and steering knuckles are so designed that when the knuckle is in the position corresponding to straight-ahead driving, the knuckle-spindle axis or wheel axis is not parallel with the plane on which the vehicle stands, but inclines downwardly from 1½° to 2°. This downward inclination is known as the camber of the axle. Camber was taken over from horse vehicle practice, in which there was a very logical reason for it. The wood wheels of horse vehicles always were strongly dished, to add to their strength; such a wheel forms a cone, bound by the steel tire at its base, rather than a disc. The most favorable conditions of stress are assured if the spokes at the bottom, which carry the load, stand vertical, and that they may do so the axle must be cambered. In motor vehicles the wheels usually are not

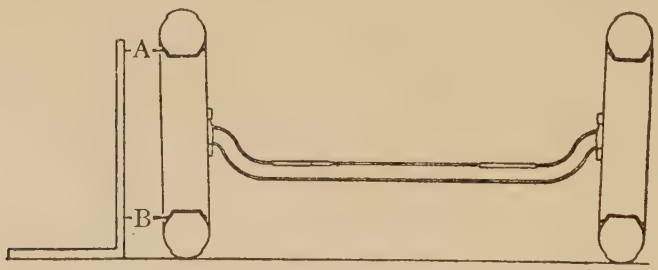


Fig. 13.—Diagram Illustrating Axle Camber.

dished, and this reason for camber therefore does not apply. The main reason for the use of camber probably is that if the spindles were set absolutely horizontal, flexing of the axle under load and any wear in the knuckle bearings would cause the wheels to spread at the bottom, which is very unsightly, as it conveys the impression of a state of advanced wear.

Motor vehicle designers in general seem to hold to the opinion that camber is desirable. The average camber is 2°. Figure 13 shows a method of measuring the camber on a completed vehicle. The difference between measurements B and A, divided by the distance between the points at which measurements are taken, is the sine of the camber angle.

Toe-in or Gather—The axes of the two steering-wheel spindles when in the position for straight-ahead running are not in a transverse vertical plane, and the front wheels are not parallel, but closer together at the front than at the rear (Fig. 14). This feature of wheel setting is known as toe-in or gather. One of the objects of toe-in, and perhaps the principal one, is to counteract certain undesirable accompaniments of camber. This may be explained as follows (see Fig. 15). When a pneumatic-tired wheel is rolled over the ground, as soon as it inclines slightly from the upright position, it no longer rolls in a straight line but in a circle.

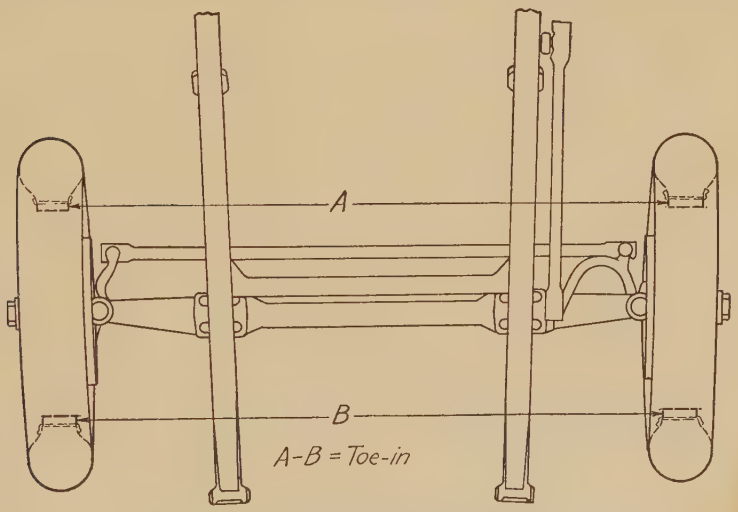


Fig. 14.—Diagram Illustrating "Toe-in."

The reason for this is that the area of contact of the tire with the road is not symmetrical with respect to the center plane of the tire; in fact, this contact area may be entirely to one side of the center plane, so that the circumference of the tire through one side of the area is considerably greater than that through the other side. The wheel tends to follow a path of least resistance, and that portion of the tread corresponding to the smaller diameter of the tire, therefore, will describe a shorter path than the portion corresponding to the larger diameter. Now, different portions of the tread can describe different paths only if the wheel rolls in a curve. This is illustrated in Fig. 15, where A is the center of the circle described by the wheel. Cambered wheels are spread apart

on top, and each wheel naturally tends to roll away from the straight path, toward the outside of the track. Since the wheels are constrained to move in a straight path by their interconnection there is constant slippage at the two sides of the tire contact area.

Now assume that the wheels have no camber but have a certain amount of toe-in. Then each wheel naturally tends to follow the direction of its plane and move inward from the straight-ahead path. Their interconnection again prevents them from doing so, and there is again slippage at the outer portions of the area of ground contact, but this time in the reverse direction, since the wheel has a tendency to turn off the straight-ahead direction in the opposite direction from that in which it tends to turn under the influence of camber. Thus, by properly proportioning camber and toe-in, their adverse effects can be made to neutralize each other. In general practice the distance between wheel rims at the

Fig. 15.—Showing How a Cambered (Inclined) Wheel Tends to Roll in a Circle.

level of the wheel axis is made 3/16 to 1/4 in. less in front than in back of the axle. The question as to the best values of camber and toe-in has been investigated by a number of tire firms in an endeavor to account for scuffing of tire treads, and James E. Hale in an S. A. E. paper reported that he had had the best results from the point of view of tire wear with a camber of 3/4° and a toe-in of 1/16 in., both of which are much smaller than the values generally used.

Trailer Effect—The steering pivots also are frequently inclined in a vertical fore-and-afte plane, as illustrated in Fig. 16. There are two reasons for this practice. The first is that the combined load due to the weight on the axle and the road resistance encountered by the wheel is in a slightly inclined direction, and in the case of an I-section axle, of course, there is an advantage in making the plane of maximum strength of the axle coincide with the direction of the

load. The other reason—and probably the more important one—is that this construction produces a trailer effect and tends to cause the steering wheels to right themselves automatically when deflected from the straight-ahead course. This effect is similar to that obtained with the front wheel of a bicycle, whereby a cyclist is enabled to ride with his hands off the handle bar. The point of wheel contact with the ground is located to the rear of the point at which the steering spindle axis produced meets the ground, hence the steering wheels trail and are automatically kept in the straight-ahead position by the road resistance. Ordinarily this trailing effect is produced by placing wedges under the front springs (W, Fig. 17).

The same effect can also be obtained by placing the axis of the knuckle slightly to the rear of the pivot axis, and this practice has been followed considerably abroad in the past, in connection with built-up knuckles. If both the knuckles and the steering gear are equipped with anti-friction bearings, and all parts are well lubricated, when the front wheels are deflected from their course and the driver then releases his hold on the wheel, they will immediately return to the straight-ahead position automatically.

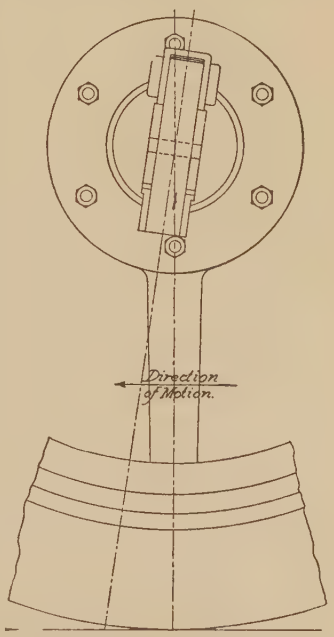


FIG. 16.—FORE-AND-AFT INCLINED STEERING PIVOT PRODUCING TRAILING EFFECT.

Spindle Diameter—As regards the diameter of the knuckle spindle, that is determined chiefly by the bore of the bearings required. The static loads on the bearings are easily determined. The two bearings together carry the whole load supported by the wheel, and the load on each bearing is in inverse proportion to the distance from its center to the center plane of the wheel. Owing to the desirability of bringing the wheel as close to the steering pivot as possible, the center plane of the wheel is usually very close to the center of the inner bearing, and this bearing carries nearly all of the

static load. The way steering-knuckle spindles are usually proportioned, on the basis of the bending moment under full static load, the factor of safety figures out to between 20 and 30. The maximum stress on the knuckle spindles, however, is due to the wheel striking a curbstone or other large obstruction at an angle. Failure due to fatigue must be guarded against, especially at the inner shoulder of the spindle, and this should always be given a liberal fillet. Since the outer bearing carries a materially smaller load than the inner one, its bore will be smaller, and the spindle therefore is stepped. The outer end of the spindle usually has a screw thread cut on it, and it is also cut with a key-way, and a washer with an integral key is threaded over it, flaps on this washer being bent over the nut to lock it in position.

Front-wheel Bearings—In normal operation the loads on the two bearings of the front-wheel spindles are practically entirely radial loads, but in turning corners at some speed the centrifugal force produces great thrust loads, for which reason combined radial and thrust bearings are usually employed for the front wheels. In the United States the major-

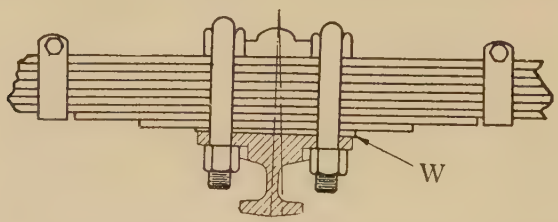


FIG. 17.—TRAILING EFFECT OBTAINED BY WEDGE UNDER FRONT SPRING.

ity of cars have taper roller bearings in the front wheels, but some are fitted with ball bearings, an angular-contact type being used.

The usual formula for centrifugal force is

$$F = 1.226\, wn^2r,$$

where w is the weight in pounds, n the number of revolutions per second and r the radius in feet. If the speed v is expressed in miles per hour the car makes

$$\frac{5280}{3600}\, v \text{ feet per second,}$$

and the circumference of the circle being $2\pi r$ feet, the car will turn at the rate of

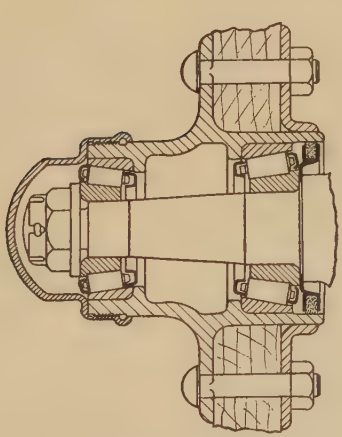
$$\frac{2\pi r}{\frac{5280}{3600}v} = \frac{r}{0.2334\,v} \text{ revolutions per second.}$$

Substituting this value for n in the expression for centrifugal force we get

$$F = \frac{0.0668\,wv^2}{r}.$$

Now let us take the case of a car with a turning radius of 20 ft., with a load of 1500 lbs. on its front axle. The turning radius of the front portion of the car will be about 17.5 ft. Let this car make its shortest turn at the rate of 15 m.p.h. The thrust load on the front-wheel bearings then will be

$$\frac{0.0068 \times 1500 \times 15 \times 15}{17.5} = 1290 \text{ lbs.}$$



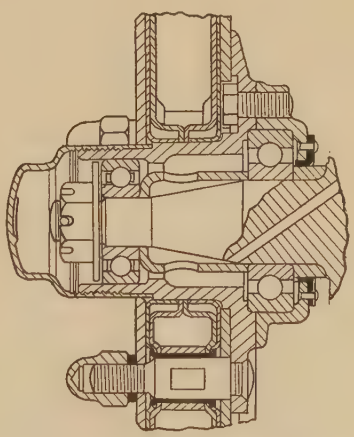


Fig. 18 (left).—Mounting of Roller Bearings on Front-Wheel Spindle.

Fig. 19 (right).—Mounting of Ball Bearings on Front-Wheel Spindle.

This thrust load will be taken chiefly by that bearing of the wheel on the outside of the curve described which takes the outward thrust. This is due to the fact that the centrifugal force removes weight from the inner wheel and adds it to that normally on the outer wheel, so that the inner wheel

will skid readily and cannot take any great amount of thrust.

In selecting bearing sizes for the front wheels it is well to consult with the bearing manufacturers, after the statical loads on both bearings have been calculated.

A typical front-wheel bearing mounting is shown in Fig. 18. The inner race of the inner bearing rests against a shoulder on the knuckle spindle, while the outer race rests against an inward flange in the wheel hub. The outer race of the outer bearing is similarly supported by an inward flange in the wheel hub, while the inner race of the outer bearing rests against a nut on the wheel spindle by means of which the two bearings can be adjusted simultaneously. The same mounting plan is followed if angular-contact-type ball bearings are used on the spindle. If both bearings are of the annular-ball type and the inner one takes both radial and thrust loads, a practice that is followed on many European light cars, the inner races of both bearings must be fixed on the spindle against endwise motion and the outer race of the larger, inner bearing is also fixed, while that of the smaller, outer bearing is left free. A hub with a bearing mounting of this description is shown in Fig. 19.

Knuckle Arms—The steering knuckles are provided with arms for interconnection of the two knuckles on opposite sides of the car by the tie-rod, and also for connection to the steering gear through the drag link. Of course, only one of the knuckles needs to be connected to the steering gear, and usually the arm of this knuckle is made double, though occasionally, especially with reverse Elliott-type steering heads, one knuckle is provided with two separate arms. The necessary length of the arms and other details will be considered in Chapter XIII on the Steering Gear. It is usually necessary to bend these arms out of the horizontal plane, at least with Elliott-type steering heads, because the tie bar has to pass underneath the body springs and the drag link must pass either over or under the axle. Moreover, with Elliott-type steering heads the arm for connection to the drag link must be given a considerable curve in the horizontal plane, because with the front wheel in the central or straight-ahead position this arm usually extends practically in the direction of the axle, hence it must curve around the vertical part of the steering head, and it must also clear this part for any position of the front wheels. Knuckle arms are secured to steering knuckles with a tapered seat, being bolted and keyed. The taper is made about 1 : 10, and the nut is secured with a cotter pin. The key is necessary because of the crank effect due to the bend in the arm. In heavy passenger cars a two-

bolt fastening is sometimes used, to reduce localization of stress.

Strength of Steering Parts—As regards the necessary strength of the knuckle arms and of all other parts of the steering linkage, that depends upon the maximum moment around the knuckle-pin axis set up by road resistance or road shock on the wheel. This moment was formerly considered to be proportional to the distance between the center plane of the wheel and the pivot axis. That assumption is correct for the case of running on smooth roads. However, with so-called center-point steering, where the pivot axis lies in the center plane of the wheel or the pivot axis produced passes through the center point of tire-ground contact, the length of this arm is nil, and the moment, consequently, also is zero, but it is well known that such a construction calls for steering members of nearly the same strength as required with the ordinary design of steering head. The reason for this is that, as already pointed out, the maximum moment around the knuckle pivot occurs when the wheel strikes the curb at an angle, or when the driver pulls hard on the steering wheel to get the wheels out of deep ruts. The force producing the moment in such cases is not in the plane of the wheel, but at a substantial angle thereto, and the moment is not proportional to the distance between the center plane of the wheel and the pivot axis; it is substantially proportional to the diameter of the wheel and, of course, also to the load supported by the wheel.

This moment around the knuckle pivot must be balanced by the moment due to the force acting at the end of the knuckle arm, which is equal to the product of this force into the length of the arm. The effective moment at the section of the knuckle arm where its tapered root emerges from its socket in the knuckle will be somewhat smaller, for the reason that the distance of this section from the eye is slightly greater than the distance of the pivot axis therefrom. However, the proportion between these two distances will be substantially the same with different designs, and we may therefore say that the resisting moment of the section of the knuckle arm where it emerges from its socket should be proportional to the maximum moment around the knuckle pivot due to road shock, which we have shown to be proportional to the load on the wheel and the wheel diameter. That is to say,

$$d^3 \sim LD,$$

where d is the diameter of the round section of the arm; L,

the maximum load on the wheel and D, the wheel diameter. Available data show that in average American practice

$$d = \sqrt[3]{\frac{LD}{28{,}000}} \text{ in.}$$

The arm proper is generally made of oval section, of such dimensions as to have the same section modulus as the large end of the tapered section near that section, and tapering down slightly toward the free end. Figure 20 shows a typical design of knuckle arm for Elliott-type steering heads. The steering arms may extend either forward or to the rear of the axle. With forwardly extending arms the tie-rod, which connects the arms of knuckles at opposite ends of the axle,

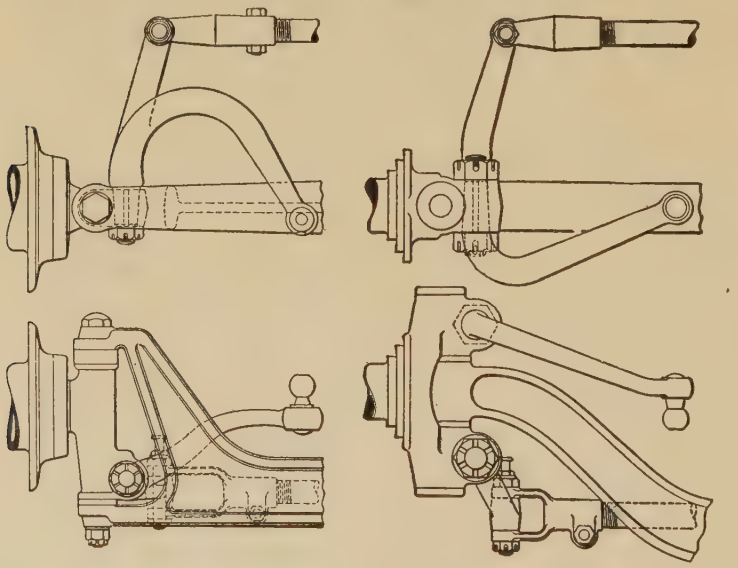


Fig. 20 (left).—Knuckle Arm for Elliott-Type Steering Head.
Fig. 21 (right).—Knuckle Arm for Reversed Elliott-Type Steering Head.

must be located in front of the axle, and for various reasons this construction is not considered as practical as where the arms extend toward the rear, which arrangement is almost universal practice. When the knuckle arms extend forwardly they must approach the wheels in order to give correct steering, and with brakes on the front wheels this is impossible,

because there is no room. Moreover, the steering tie-rod is better protected when located back of the axle. The knuckle arm shown in Fig. 20 extends toward the rear. Figure 21 shows a knuckle arm for a reverse Elliott axle end and knuckle. One of the advantages of this type of steering head is that it facilitates the fastening of the knuckle arms. With a reverse Elliott-type steering head the knuckle on the side where the steering gear is located usually has two arms fastened to it, one for the tie-rod and the other for the drag link.

Tie-rod—The rod which connects the arms of knuckles at opposite ends of the axle is now generally made tubular. Whether located in front of or behind the axle, it is subjected alternately to tension and compression, but it is subjected to the greatest stresses by the compression, which makes it plain why a tubular section is preferable, as the rod then acts as a strut or column, whose strength depends upon the least radius of gyration of the section.

Considering the rod located back of the axle, the maximum compression load may be represented by the expression

$$P = \frac{cWD_w}{b}.$$

where c is a constant; W, the maximum load on the wheel; D_w, the wheel diameter and b, the length of the knuckle arm. The rod acts like a column with free ends, the permissible load for which is

$$F = \frac{SA}{1 + q\frac{l^2}{r^2}}.$$

(Rankin's equation), where S is the safe working stress of the material; A, the sectional area of the rod; q, a constant; l, the length of the rod, and r the least radius of gyration of the section. For a solid rod,

$$r^2 = \frac{D^2}{16}$$

for a tube,

$$r^2 = \frac{D^2 + d^2}{16}.$$

Hence

$$\frac{cWD_w}{b} = \frac{SA}{1 + q\frac{l^2}{r^2}}.$$

With the proportions obtaining in steering tie-rods we may write

$$\frac{cWD_w}{b} = \frac{SA}{\frac{l^2}{q\,r^2}}$$

without committing a great error. Remembering that for a hollow circular section

$$A = \frac{\pi}{4}(D^2 - d^2)$$

and

$$r^2 = \frac{D^2 + d^2}{16},$$

we get, by substitution,

$$\frac{cWD_w}{b} = \frac{S\pi\,(D^4 - d^4)}{64ql^2},$$

which may be transformed to read

$$\left(\frac{S\pi}{64qc}\right)(D^4 - d^4) = \frac{WD_w l^2}{b}.$$

All of the factors in the first pair of parentheses may be regarded as constant, and we may denote this term by C, which gives us

$$D^4 - d^4 = \frac{Wl^2 D_w}{Cb}.$$

The value of C should be chosen at from 25,000,000 to 30,000,000. In using the last equation, calculate the value of the right-hand term, assume a value for D and calculate the necessary value of d. If the result is unsatisfactory, assume some other value of D. The wall thickness must not be too small, because the wall will have to be threaded for the connector (unless some other means of fastening is employed) and the thread must not cut down the strength of the wall too much. The above equation is intended for straight rods with central connectors only. If the rod is formed with a curve at the center to clear the engine, or if offset connectors are used for the same purpose, a heavier section must be used and the value of the constant C decreased.

Tie-rod Connectors—If the knuckle pivot axes at opposite ends of the axle are vertical (and therefore parallel), the ends of the knuckle arms swing in the same plane and

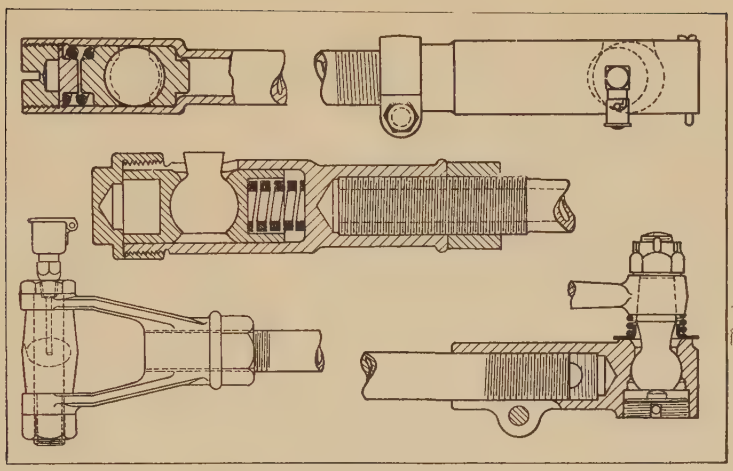


FIG. 22.—VARIOUS DESIGNS OF TIE-ROD CONNECTOR.

can be connected by plain forked connectors. On the other hand, if the pivot axes are not vertical, the ends of the knuckle arms swing in different planes and the connectors must be of a universal type. A number of connectors of both types are illustrated in Fig. 22. The upper view shows connectors used at opposite ends of the tie-rod, that on the left-hand end being integral with the rod, while the one on the right end is screwed over the rod and therefore provides for adjustment. In some of the designs a lock nut is used

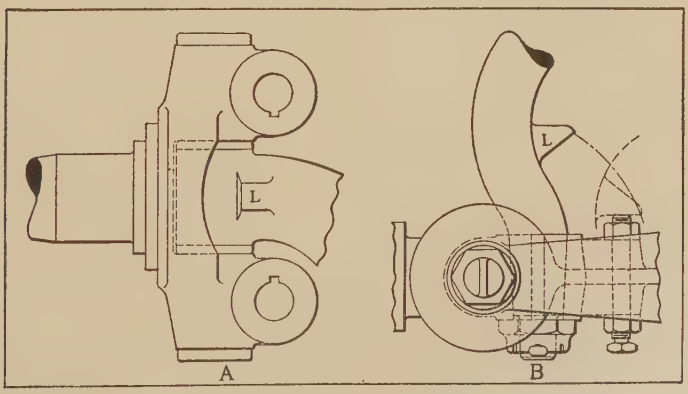


FIG. 23.—STEERING STOPS.

to take up any slack there may be in the thread, while in other cases the connector is slotted and clamped on the tie-rod by a clamp bolt. Two of the ball-and-socket connectors have one of the sockets spring-cushioned, which automatically takes up any wear. The connector shown in the lower right-hand corner is of the safety type, in that there is no possibility of the ball joint coming apart. All of the connectors have lubricating fittings. In many cases separate ball studs are used with these connectors. These have been standardized by the S. A. E. (see Appendix). The ball diameter is usually chosen slightly larger than the outside diameter of the tie-rod proper.

Steering Stops—To prevent contact between the tire and the car frame or the drag link of the steering mechanism when the front wheels are swung all the way over to one side, the front axle is provided with a so-called steering stop. Such a stop (L) is most easily provided in the case of a reversed Elliott-type steering gear, as shown in Fig. 23 at A. At B in the same figure is shown the arrangement used in connection with Elliott-type front axles. In this case a lug L on the knuckle arm contacting with an adjustable stop on the axle forging limits the steering motion.

CHAPTER XIII

The Steering Gear

Historical—Instead of the fifth-wheel steering arrangement used on horse vehicles, the divided axle is universally employed on automobiles. This was invented by Lankensperger, of Munich, in 1817. The English patent on it was taken out in the name of Rudolph Ackermann, and in English-speaking countries the gear, in consequence, has come to be known as the Ackermann steering gear. A refinement of this steering mechanism for automobile purposes was introduced in 1878 by Charles Jeantaud, a French carriage builder, who devised what is known as the Jeantaud diagram. Jeantaud, it seems, recognized the principle that if the vehicle is to turn a corner without sideward slip of any of the wheels, the linkage of the steering wheels must be so arranged that the axles of all the wheels produced always intersect a common vertical line, the latter forming the momentary axis of rotation. Jeantaud found that in order to approximately fulfil this condition, the steering arms, instead of being parallel, must be inclined toward each other when they extend to the rear of the axle, and away from each other when they extend forward of the axle; and his diagram, which is intended to give the correct inclination of the arms, indicates that the center lines of the arms produced should meet at the middle of the rear axle. Later investigations of the steering problem showed that with the ordinary trapeze form of steering linkage it is impossible to absolutely satisfy the condition of correct steering, and that for a minimum error for the whole steering range, with the proportions of wheelbase to tread found in modern cars, the point of intersection of the two steering arms produced lies some distance in front of the rear axle. With the very short wheelbases employed during the early years of the automobile movement, together with a standard tread, Jeantaud's diagram was substantially correct.

Theory of Steering Mechanism—In the following investigation we will denote the length of the wheel base by W; the distance between steering pivots by L; the inclination of the inner wheel axis by α; the inclination of the outer wheel axis by β; the inclination of the knuckle arms by θ; the length of the arms by l.

Referring to Fig. 1, it will be seen that

$$\frac{ac}{cd} = \cot \alpha$$

and

$$\frac{bc}{cd} = \cot \beta.$$

Hence

$$\frac{bc - ac}{cd}\left(= \frac{L}{W}\right) = \cot \beta - \cot \alpha.$$

This equation enables us to plot the required values of β corresponding to different values of α for any ratio L/W. It expresses the condition which should be satisfied by the gear, but furnishes no guide as to how this may be accomplished.

Graphical Solution of Steering Problem—There is no direct analytical method for determining the most advantageous

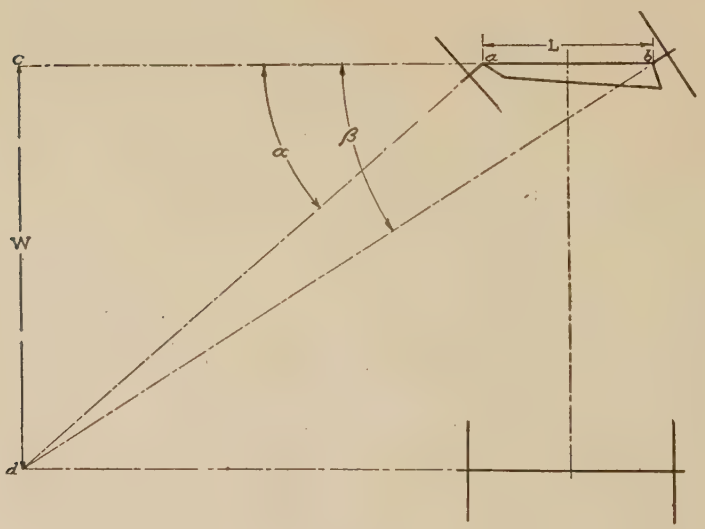


Fig. 1.

angle of the knuckle arms, and some graphical method is usually employed. By laying the steering diagram off on the drawing board to, say, half size, a sufficient degree of accuracy is attained. Unfortunately, for small deflections of the front wheels, the distance of the point of intersection of the wheel axes is so far from the axis of the car that it falls far outside the limits of an ordinary drawing board, and accuracy of the linkage with small deflections is of special importance, for the reason that the wheels are turned through a small angle very much oftener than through a big angle, and the car generally runs at a much higher speed when describing curves of large than of small radius. This difficulty may be overcome as follows (Fig. 2): From points a and b, denoting the steering pivots, draw lines perpendicular to the axles which will intersect the rear axle at e and f. Next, draw a line from the middle point g of the front axle to point e. Then lines drawn from the pivot points a and b to any point on line ge will make with the front axle corresponding steering angles. This may be proven as follows:

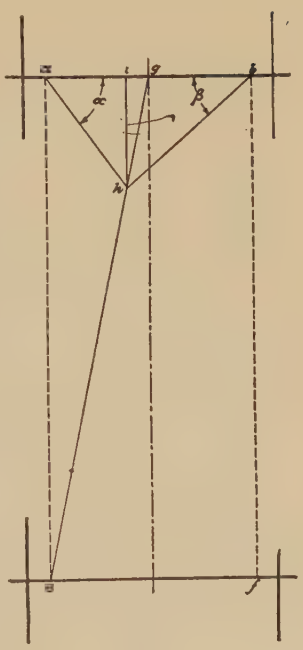


FIG. 2.

$$\cot \alpha = \frac{ai}{ih} = \frac{ag - ig}{ih},$$

and

$$\cot \beta = \frac{bi}{ih} = \frac{ag + ig}{ih}$$

$$\therefore \cot \beta - \cot \alpha = \frac{2ig}{ih}.$$

But

$$\frac{ig}{ih} = \frac{ag}{ae},$$

and substituting,

$$\cot \beta - \cot \alpha = \frac{2ag}{ae} = \frac{L}{W}.$$

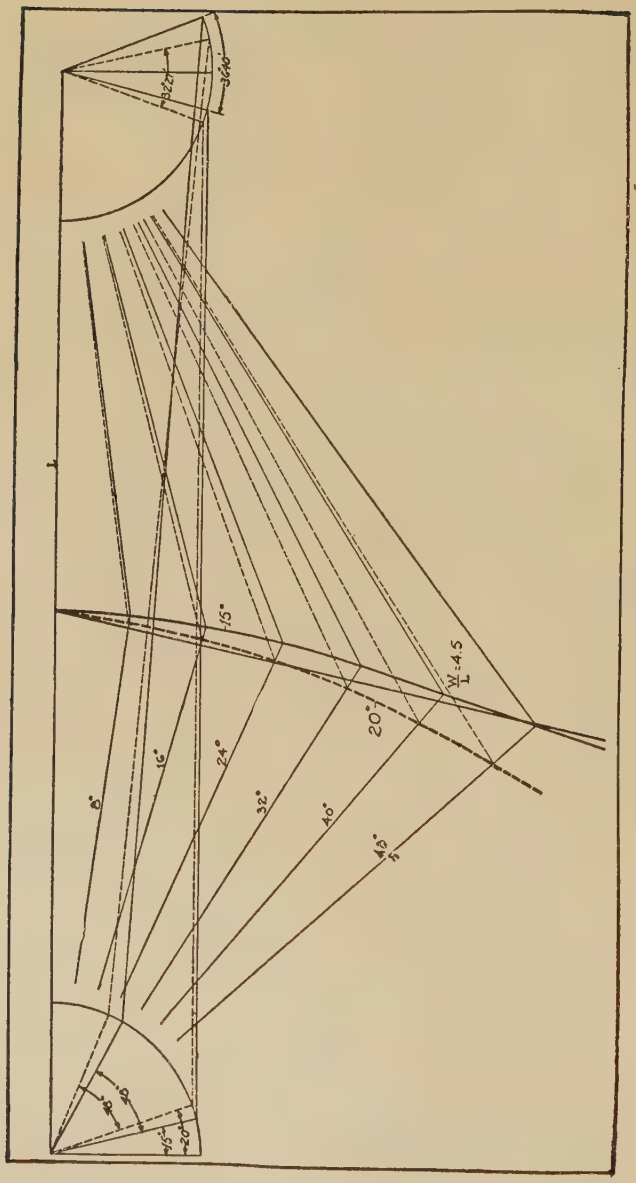


FIG. 3.

Now assume steering arms of a definite length and making a certain angle with the longitudinal axis of the vehicle. Next determine graphically the deflection of the outer wheel for various assumed deflections of the inner wheel, say, 8°, 16°, 24°, 32°, 40° and 48°, for these steering arms. This has been done in Fig. 3 for two particular cases. The following dimensions were assumed in making this drawing: $L = 50$ in.; $l = 7$ in. and angle $\Theta = 15°$ and 20°, respectively. The values of angle β were determined graphically for values of angle α of 8°, 16°, 24°, 32°, 40° and 48°, respectively. After these values of angle β had been found, corresponding angles α and β were laid off on opposite ends of the line L. Through the points of intersection of the lines describing corresponding angles were drawn curves, one for 15° and the other for the 20° knuckle arms. These may be called steering error curves, because their deviation from the diagonal line *ge* (Fig. 2) indicates the error in the steering angles. In Fig. 3 the diagonal *ge* corresponding to the value $L/W = 0.45$ is drawn in. It will be seen that for small deflections the angle of the outer wheel is too large with both 15° and 20° knuckle arms. The 20° knuckle arm gives the correct deflection of the outer wheel at about 26° of the inner wheel, and the 15° knuckle arm gives the correct deflection of the outer wheel at 46° deflection of the inner wheel. Beyond these points the angle of the outer wheel is too small. It may readily be seen from this that the most advantageous angle of the knuckle arms depends upon the turning range of the inner wheel. Thus, if the motion of the inner wheel were limited to 32°, the 20° knuckle arm would be the best, whereas if the range of motion of the inner wheel were as large as 45°, the angle of the knuckle arm should be about 18°—for a value of $L/W = 0.45$.

In order to use this method for the practical determination of the proper knuckle arm angle, steering error curves for different knuckle arm angles and lengths should be laid out very carefully for permanent use, and in any particular case the diagonal *ge* corresponding to the particular value of L/W should be placed on the chart in pencil, when the most advantageous knuckle arm angle will at once be apparent.

An Analytical Method—An ingenious analytical method of determining the deflection of the outer wheel corresponding to a given deflection of the inner wheel was developed by Herbert C. Snow and is briefly summarized in the following:

Four different cases have to be considered, viz., with the knuckle arms in front and in the rear, respectively, and with

the knuckle arm angle θ greater and less than the deflection β of the outer wheel, respectively. In Fig. 4 the knuckle arms extend to the rear of the axle and angle θ is greater than β. In this diagram the knuckle arms are purposely shown abnormally long, for the sake of greater clearness. Referring to the drawing,

$$M = L - 2l \sin \theta \qquad ge = l \sin (\alpha + \theta)$$

$$ga = l \cos (\alpha + \theta) \qquad hi = jb = l \sin (\theta - \beta)$$

$$jf = l \cos (\theta - \beta) \qquad ge + M - N + hi = L$$

Substituting values of ge and hi,

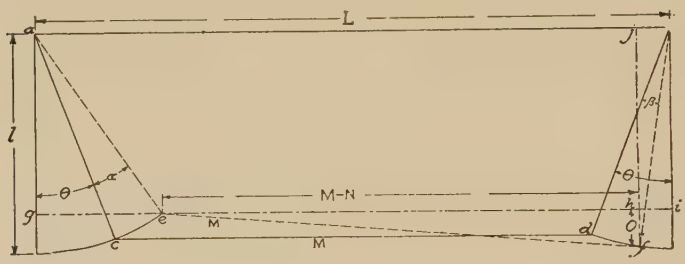


FIG. 4.

$$l \sin (\alpha + \theta) + M - N + l \sin (\theta - \beta) = L$$

Substituting the value of M,

$$l \sin (\alpha + \theta) + L - 2l \sin \theta - N + l \sin (\theta - \beta) = L.$$

Transposing and dividing by l,

$$\sin (\theta - \beta) = 2 \sin \theta - \sin (\alpha + \theta) + \frac{N}{l} \quad . \quad . \quad . \quad (1)$$

By a similar process of reasoning it is found that with β greater than and the arms extending to the rear

$$\sin (\beta - \theta) = \sin (\alpha + \theta) - 2 \sin \theta - \frac{N}{l} \quad . \quad . \quad . \quad (2)$$

with β smaller than θ and the arms in front of the axle—

$$\sin (\theta - \beta) = 2 \sin \theta - \sin (\theta + \alpha) - \frac{N}{l} \quad . \quad . \quad . \quad (3)$$

and with β greater than θ and the arms in front of the axle

$$\sin(\beta - \theta) = \sin(\theta + \alpha) - 2\sin\theta + \frac{N}{l} \quad . \quad . \quad . \quad (4)$$

These various equations cannot as yet be solved, because N is not known. The value of N in terms of known factors may be found as follows:

$$O = jf - jh.$$

When β is smaller than θ,

$$jf = l\cos(\theta - \beta)$$

hence

$$O = l\cos(\Theta - \beta) - l\cos(\alpha + \Theta)$$
$$= l[\cos(\Theta - \beta) - \cos(\alpha + \Theta)] \quad . \quad . \quad (5)$$

Similarly, when β is greater than θ,

$$jf = l\cos(\beta - \Theta)$$

and

$$O = l[\cos(\beta - \Theta) - \cos(\alpha + \Theta)] \quad . \quad . \quad . \quad . \quad . \quad (6)$$

In every case

$$M - N = \sqrt{M^2 - O^2}$$

and

$$N = M - \sqrt{M^2 - O^2} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (7)$$

The equations thus derived permit of accurately determining the angle of the outer wheel corresponding to any angle of the inner wheel, the proportions of the linkage being given. The following example shows its method of application: Suppose the distance L between pivots to be 50 in.; the length l of the knuckle arms, 6.5 in.; θ, 20°, and α, 30°, the knuckle arms extending to the rear of the axle. Then according to equation (2)

$$\sin(\beta - 20°) = \sin(20 + 30)° - 2\sin 20° - \frac{N}{l}$$

Disregarding the term N/l for the moment we get for a first trial value

$$\sin(\beta - 20°) = 0.082.$$

$$\beta = 24° 42'.$$

Inserting this value of β in equation (6) we get

$$O = 6.5\ (0.9966 - 0.6428) = 2.2997,$$

$$M = 50 - (2 \times 6.5 \times 0.342) = 45.56$$

and inserting these values in equation (7) we get

$$N = 0.053.$$

Now using this value N in equation (2) we get

$$\beta = 25^\circ\ 10'.$$

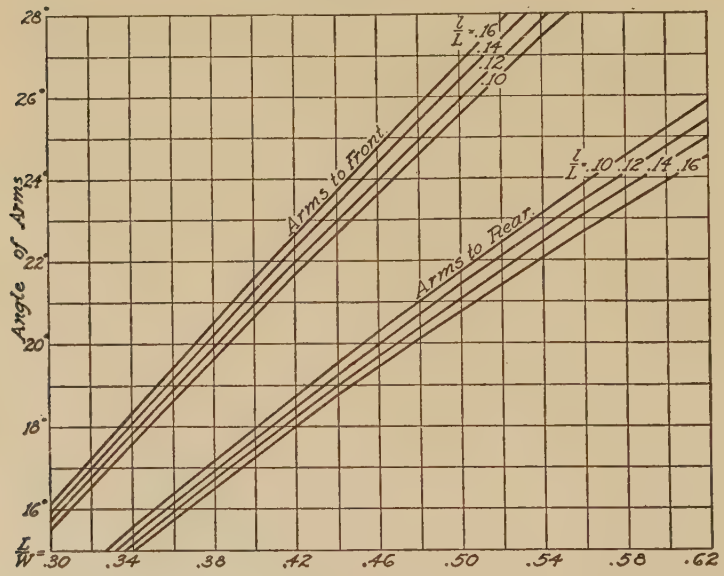


Fig. 5.—Proper Knuckle-Arm Angles for a Limiting Deflection of 33°.

The calculations could be continued further, but Mr. Snow has shown that the second trial value is correct within one minute in the most extreme case, which is as high a degree of accuracy as is required in practical work.

Using this method, Mr. Snow calculated the error in the steering angle for each of the turning angles 25° and 30° of the inner wheel, with the knuckle arms set at from 15° to 30°, extending both to the front and the rear, using four ratios l/L in each case, viz., 0.10, 0.12, 0.14 and 0.16. The results of these calculations are plotted in Figs. 5 and 6. These

diagrams give the best values for the angles of the knuckle arms for limiting turning angles of 33° and 40° of the outer wheel, when the values of L/W and l/L are known.

General Arrangement of Steering Gear—Motor vehicles are steered by means of hand wheels, the wheel being secured to the upper end of a steering wheel shaft which at its lower end carries the steering gear proper. The steering wheel is usually 17 in. in diameter in the lighter passenger cars, 18 in. in the larger passenger cars and the lighter trucks and buses, and 20 and even 22 in. in the heaviest trucks. In most cases the engine-control levers are mounted on top of the steering

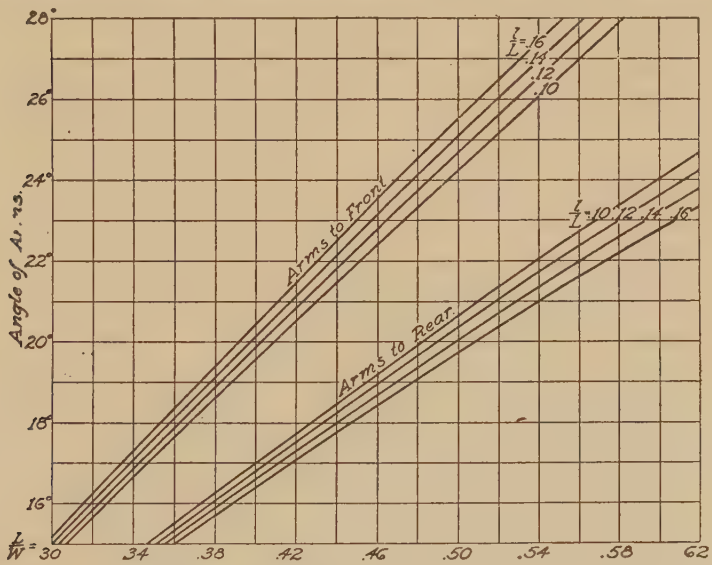


FIG. 6.—PROPER KNUCKLE-ARM ANGLES FOR A LIMITING DEFLECTION OF 40°.

wheel, which necessitates making its shaft tubular so that other tubes (and perhaps electric cables for the electric horn) can pass through it. On the outside the steering-wheel shaft generally is surrounded by a non-rotating light tube or jacket. This entire assembly of concentric tubes is referred to as the steering column.

The steering gear, which reduces the angular motion of the steering wheel, is located at the bottom of the column in a cast housing which is supported on the vehicle frame (or the engine) by means of a suitable bracket. A shaft pro-

truding from this housing carries the steering gear arm, from which connection is made to one of the steering knuckles through a drag link with universal joints, usually of the ball-and-socket type. A general assembly view of the whole steering mechanism is shown in Fig. 7.

In most cases the steering column is mounted on the car at an angle, for comfort in driving. This angle varies a good deal in different classes of vehicle. In motor trucks and buses, where it is desirable to economize space, the column is sometimes made nearly vertical, while in so-called speedsters or sport models the inclination from the vertical may be as much as 60°. These vehicles have very low seats, and a sharp steering column rake is necessary to permit of

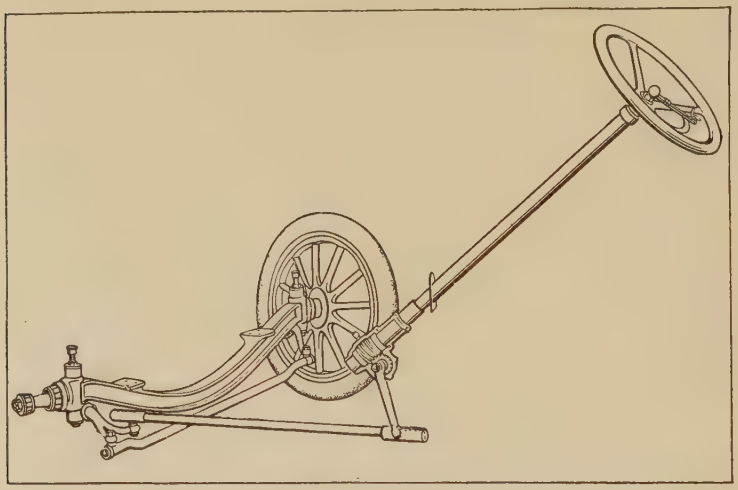


Fig. 7.—General Assembly View of Steering Mechanism.

steering in comfort. On stock passenger cars an inclination of 50° to the vertical is much used.

Right vs. Left Steering—For a long time there was much difference of opinion as to whether the steering column could be mounted to greater advantage on the right or the left side of the car. With the American rule requiring all traffic to keep to the right side of the road, a driver can better gage his distance from vehicles coming in the opposite direction if he sits on the left, but he can better judge his distance from the curb or from a vehicle that is being overtaken by him, if he is seated on the right. There are other points in favor of each position, but as the result of long experience

American engineers have come to the conclusion that with the right-hand rule of the road it is better to mount the steering gear on the left and drive the car from that side. All American cars for domestic consumption are now so built.

In England and in some of the smaller Continental countries drivers keep to the left of the road, and it is obvious that if left-hand steering is correct in a country in which traffic keeps to the right, right-hand steering is correct in a country where it keeps to the left. Practically all cars manufactured in England have the steering gear mounted on the right-hand side. France and Germany follow the right-hand rule of the road, but practice with respect to the location of the steering gear is not standardized in these countries; it seems to be evolving in the direction of left-mounting, however.

Steering-Gear Requirements—The requirements made of a steering gear are somewhat conflicting, and, like most other engineering products, the average steering gear is a compromise. The primary object of the gear is to multiply the turning effort produced on the wheel by the driver, so that he can steer the vehicle without exerting too much physical effort. Then, the steering gear should be to a certain degree irreversible or back-locking, so that shocks sustained by the road wheels will not be transmitted in toto to the driver's hands. These two requirements already conflict in a degree, for the more friction there is in the steering gear, the better the shocks are absorbed, while, on the other hand, with a great deal of friction in the gear it is difficult to operate it. Fortunately, there are mechanisms whose efficiency is much greater when motion is transmitted in one direction than when it is transmitted in the other, and this difficulty, therefore, can be circumvented. However, it is often desired to have a trailing or self-righting effect, so that if the road wheels are slightly deflected from the straight-ahead course and the driver then releases the wheel, the road wheels automatically return to the straight-ahead position. Such an effect can be obtained only if the resistance to the transmission of motion from the road wheels to the steering wheel is comparatively low.

Types of Steering Gear—Two fundamental types of mechanism are used for reducing the motion of the steering wheel, the worm and wheel (or worm-wheel sector) and the screw and nut. There are a good many designs of steering gear in use today which could not be absolutely classified as one or the other of these two, but practically all are derived

from one or the other. The motion at the contact surfaces is entirely sliding in the screw and nut, and chiefly sliding in the worm and wheel, and with both the efficiency of transmission is far less for one than for the other direction.

Steering gears are now made with ratios ranging from 11 or 12 to 1 for light cars to about 18 to 1 for heavy passenger cars and even somewhat greater for heavy buses and trucks. These reductions are almost twice as great as were used during the earlier period of the automobile industry. Adoption of balloon tires, with their greater ground-contact area; the need for frequent manoeuvering in close quarters in parking, and the fact that many cars are now being driven by women make these larger reductions desirable. If the steering arm has a range of angular motion of 70°, this implies 2⅓ to 3½ complete turns of the steering wheel in moving the road wheels from hard over one way to hard over the other way.

Dimensioning of Steering Gear Parts—It was pointed out in Chapter XII on The Front Axle that the maximum moment around the knuckle pin axis, which determines the stresses imposed on all steering parts, is proportional to the weight carried by the front axle and the diameter of the front wheel. By the drag link this moment is transferred to the steering arm shaft, but in being transferred it is increased or decreased in the proportion l_1/l_2, where l_1 is the effective length of the steering-gear arm and l_2 that of the knuckle arm. This ratio, as a rule, is close to unity. The torsional resisting moment of the steering-gear shaft is proportional to the cube of its diameter and to the shearing strength of the material of which the shaft is made. Hence we may write,

$$d^3S \sim WD,$$

where d is the diameter of the steering-gear shaft; S, the shearing strength of the material of which it is made; W, the maximum load on the front axle, and D, the diameter of the front wheels. The factor of safety must be somewhat greater in high-speed vehicles than in low-speed trucks. The author has found that where carbon steel is used for the steering-gear shaft (from 1020 to 1045) its diameter is made equal to

$$d = \sqrt[3]{\frac{WD}{36{,}000}} \text{ in.}$$

With alloy steel the denominator of the fraction under the

radical sign can be made larger in the proportion of the shearing strengths. With steels Nos. 2330 and 3140 it can be made 54,000, and for slow-speed truck work as high as 70,000. This assumes that the steering arm is secured to the shaft by a serrated joint in accordance with the S.A.E. Recommended Practice (See Appendix), which does not greatly reduce the strength of the section.

Dimensions of other parts of the steering gear can be based on that of the steering-gear shaft. For instance, the torque capacities of both a shaft and a worm and worm gear are proportional to the cubes of the linear dimensions. The basic dimension of a worm-and-wheel set may be taken to be the center distance, and this, therefore, should be proportional to the shaft diameter. In American practice the center distance is made equal to twice the shaft diameter or very slightly more, wheeras in European practice it is frequently three times the shaft diameter. Apparently a center distance equal to twice the shaft diameter—if the shaft is proportioned in accordance with the equation given in the foregoing—gives all the strength needed, but with the larger center distance the pressure on the teeth will be less, hence the wear will be less, and, besides, a certain amount of wear on the teeth will cause less backlash in the steering wheel.

Having determined the approximate center distance required, the elements of the worm and wheel can be calculated in accordance with the rules given in Chapter VIII on The Worm Drive. For instance, the ratio of the number of teeth in the wheel to the number of leads on the worm must be equal to the steering ratio desired. Formerly the worms often had three leads, but with the increased steering ratios now in use, two is the largest number, and with certain types of modified worm gears a single thread is used.

Worm and Wheel vs. Worm and Sector—Since the steering arm and its shaft rotate through only about 70°, it is not necessary to use a complete wheel with the worm. In fact, worm-and-sector steering mechanisms were used earlier than the worm-and-wheel-type and are still widely used in Continental Europe. The reason for using a complete wheel instead of a sector is that there is no satisfactory way of making adjustments for wear on the teeth of worm gearing. By using a complete wheel, after the teeth on one section of the wheel have worn so much that the backlash has become objectionable, the steering arm may be moved through an angle of 90° on its shaft, and thus an entirely new set of teeth brought into action. Thus the tooth surfaces on the wheel can be renewed three times, and for a certain life the

gear can be made smaller. In a worm-and-sector-type of steering gear the center distance should be made at least 2.5 times the calculated shaft diameter for carbon steel.

While there is no satisfactory adjustment for wear in worm gear, an eccentric adjustment is sometimes provided on the wheel shaft or sector shaft to compensate for inaccuracies in the machining of the housing. The shaft is mounted in an eccentric bushing that is provided with an arm whose outer end can be secured to the frame in different positions, thus permitting of angular movement of the eccentric bushing.

In the smaller cars the steering-wheel shaft is sometimes

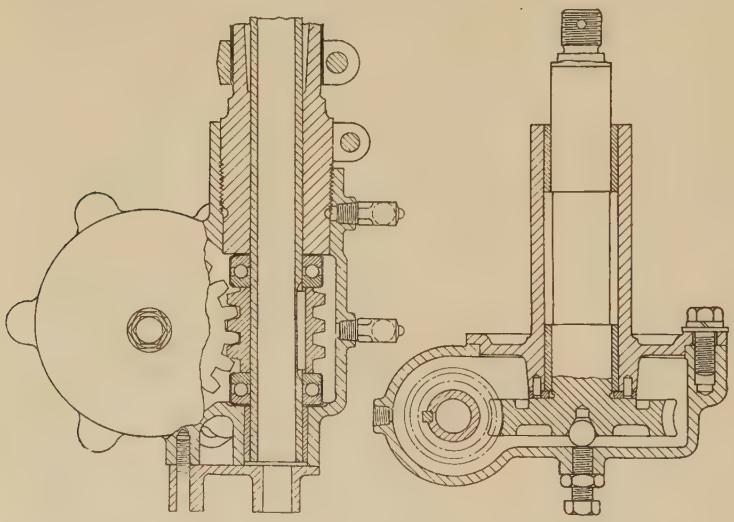


FIG. 8.—TYPICAL WORM-AND-WHEEL STEERING GEAR.

made solid, the controls being placed on the outside of the column. In that case the shaft is made either ¾ or ⅞ in. in diameter. Hollow steering-wheel shafts, which are the rule, are made with outside diameters of from 1 to 1¼ in., and the jackets surrounding them are from ⅜ to ¾ in. larger in diameter.

Worm-and-Wheel Steering Gear—Figure 8 shows a typical worm-and-wheel steering gear. The worm is keyed to the lower end of the tubular steering-wheel shaft. Radial loads due to worm-tooth contact are taken up on two plain bearings in the case, while thrust on the worm in both direc-

tions is taken up on ball thrust bearings. The upper one of the two plain bearings is a steel bushing screwing into the steering-gear housing, and in addition to acting as bearing for the steering-wheel shaft, it serves as an adjusting means for the ball thrust bearings and a lower support for the steering column jacket. A lug is formed on the upper end of the boss on the housing into which this bushing screws, which lug is split and can be tightened on the bushing by means of a clamp bolt.

The worm wheel is formed integral with its shaft. Radial

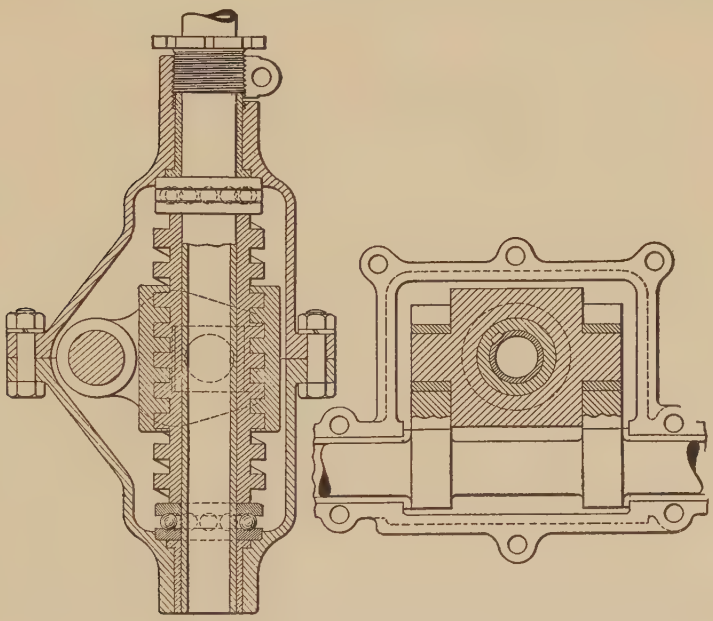


Fig. 9.—Screw-and-Nut Steering Gear.

loads on it are taken up on two bearings on the same side of the worm, in a long hub on the housing, while thrust loads on the wheel are taken up on a single central ball in one direction and a thrust washer in the other. A set screw with lock nut serves to take up any slack in the direction of the worm-wheel axis.

Screw-and-Nut Steering Gear—A screw-and-nut steering gear is illustrated in Fig. 9. The screw has square threads and is surrounded by a correspondingly threaded nut. This

latter has trunnions formed on its outside, which carry square trunnion blocks that are located in forked arms formed integral with the steering-gear shaft. In the design here shown, the nut is of square section and fits snugly between the two forked arms, which prevent it from turning with the screw. In some other designs the nut is provided with a fin or key which moves in a slot cut in the housing. In another variation the nut, instead of being provided with trunnions on opposite sides, has cylindrical recesses formed on both sides. Into these machined recesses are set cylindrical blocks with a square slot cut across their face, into which extends an arm on the

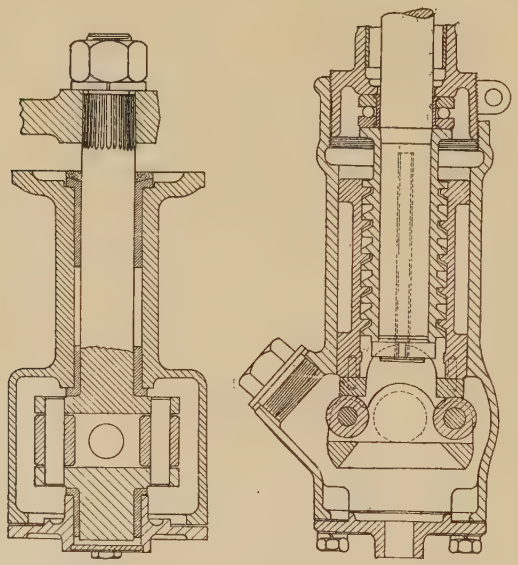


FIG. 10.—SPLIT-NUT-TYPE OF STEERING GEAR.

steering-gear shaft. A part of the side wall of the cylindrical recess is cut away to allow the arm to enter and give it freedom of motion.

A peculiarity of the screw-and-nut-type of steering gear is that its reduction ratio is not constant but varies from a minimum in the central position to a maximum at both extremes. The screw and nut of the gear illustrated are both made of steel. In some cases the nut is lined with babbitt metal forming the thread, but it must then be made somewhat longer. In order to give the babbitt a firm hold on the steel

shell, holes are drilled through the latter before the babbitt is poured.

Split-Nut-Type of Steering Gear—Backlash in steering gears is very objectionable, and numerous schemes of adjustment have been tried, but few have proven practical. Probably the most successful is that embodying the use of a screw with both a right-hand and a left-hand thread. Such a gear, as used by Packard, is illustrated in Fig. 10. The threads are of peculiar form, being square on the working face and inclined on the other. Each thread engages a correspondingly threaded half-nut. When the steering wheel is turned, one

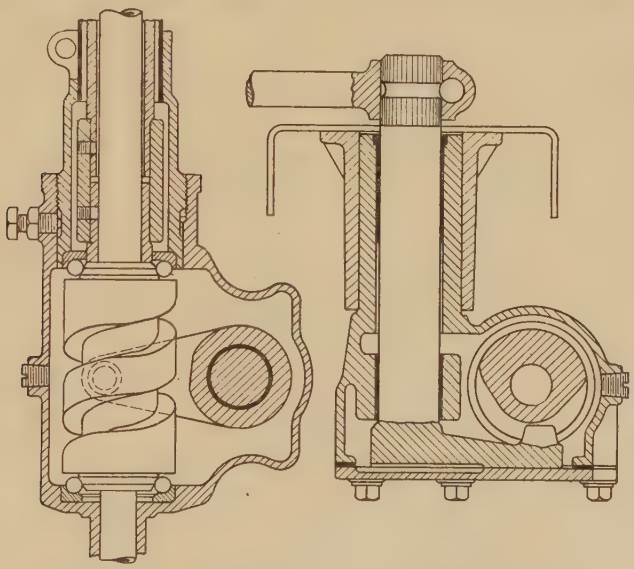


FIG. 11.—ROSS CAM-AND-LEVER STEERING GEAR.

half-nut will move up and the other down. Both half-nuts press on rollers carried by a double-armed lever formed integral with the steering-gear shaft, through contact shoes of hardened steel secured to them. With this design only a single thrust bearing is required, on top of the screw, and by forcing this down by means of the adjusting nut, any slack can be taken up. This scheme of double threads also has been applied in several different ways. For instance, it is possible to use a hollow screw, with one thread on the outside and another on the inside. Or, one outside thread may be above the other on the same shank.

Cam and Lever Steering Gear—Since the advent of balloon tires, designers have endeavored to reduce the effort required to operate steering gears. If there is a large surface in contact, as in the ordinary screw-and-nut type, and the weather is cold so that the lubricant is stiff, there is a large amount of friction to be overcome. The friction can be greatly reduced by reducing the contact area. In the Ross cam-and-lever steering gear, illustrated in Fig. 11, the contact is theoretically along a line. An arm on the steering-gear shaft carries at its end a pin with curved, tapering surfaces, which engages into a helical cam groove of variable pitch. The object in making the cam groove of variable pitch is to secure

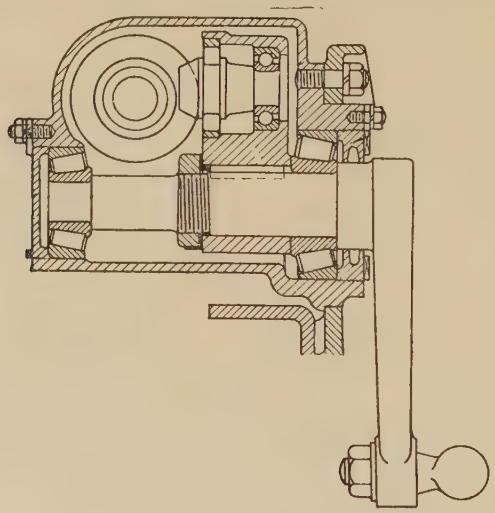


FIG. 12.—DAIMLER CAM-AND-LEVER STEERING GEAR.

a variable ratio which is smaller for the extreme positions than for the central one. Thus the steering gear will be easy to move when in the central position or near it, and still it will not take a very large angular movement of the steering wheel to move the road wheels from one extreme of their range to the other. This, of course, involves that the gear is harder to move when near either extreme position, which positions are reached particularly in maneuvering with the object of parking the car. Owing to the smallness of the contact area there is probably an extra damping action when

one of the front wheels sustains a shock, as the oil may be squeezed out.

A somewhat similar gear to the one just described is made in England by the Daimler Motor Company. In that case, as shown in Fig. 12, the pin engaging into the cam groove is at the end of a spindle mounted in ball-and-roller bearings. Hence the motion at the contact surfaces is largely a rolling one.

The Gemmer Mfg. Co. manufactures a number of different

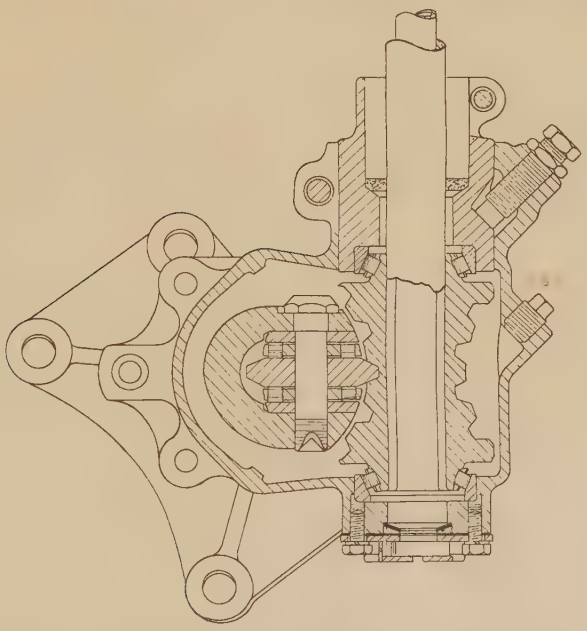


Fig. 13.—Gemmer Cam-and-Lever Steering Gear.

designs of steering gear. The worm is generally of the hour-glass type and the sector has only two teeth, or else its place is taken by a lever carrying a disc mounted on anti-friction bearings, which engages into the tooth space of the worm. In the design shown in Fig. 13, the bushing through which the steering-wheel shaft enters the housing is not screwed into the housing but has a sliding fit in its boss and is adjusted by means of a set screw and then locked by a clamp nut. This is a common feature.

Foreshortened Steering Column—In the conventional motor vehicle the steering column extends through the toe-board or dash into the space between the engine and frame, and it often interferes with the arrangement of the engine accessories. For this reason some manufacturers have adopted arrangements by which the steering column does not enter

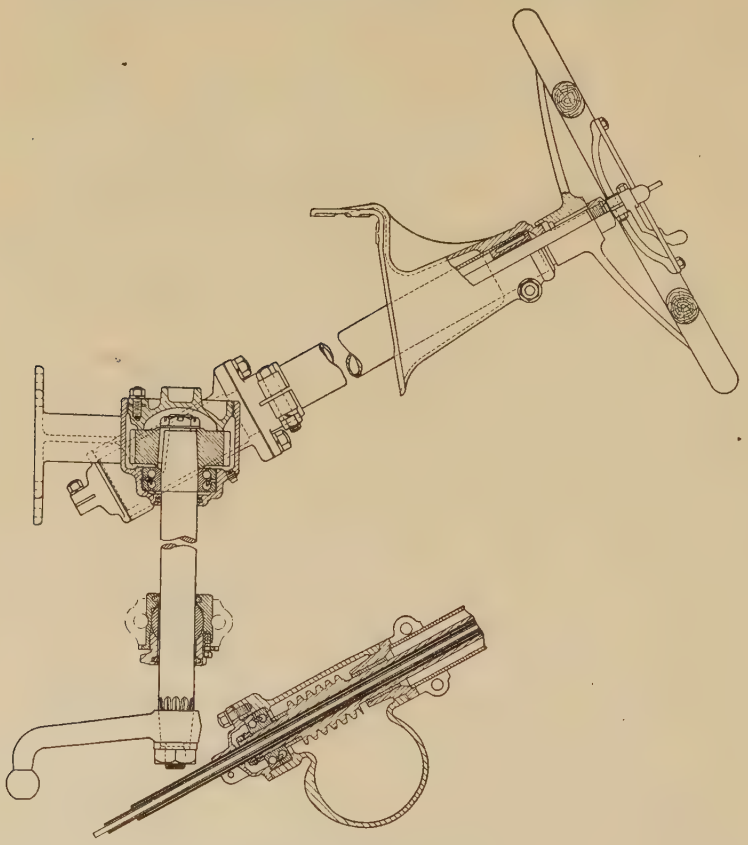


Fig. 14.—Foreshortened Steering Column.

the engine space at all. In the design shown in Fig. 14, due to the International Harvester Company, the steering-wheel shaft at its lower end just back of the dash carries a helical gear meshing with another such gear at the top of a vertical shaft, which carries the steering-gear arm at its lower end.

A spherical bearing mounted on the frame side rail steadies the vertical shaft near its lower end.

In the steering gear designed with the same object in view by the Daimler Motor Co. of England, the arrangement of the gear is conventional except that the column is much shorter and the motion is transmitted from the steering-gear arm to a bell-crank fulcrumed on the chassis frame below it, and thence forward to the knuckle arm.

Mounting of Steering Gears—The housings of the gears are generally made of malleable iron and are provided with

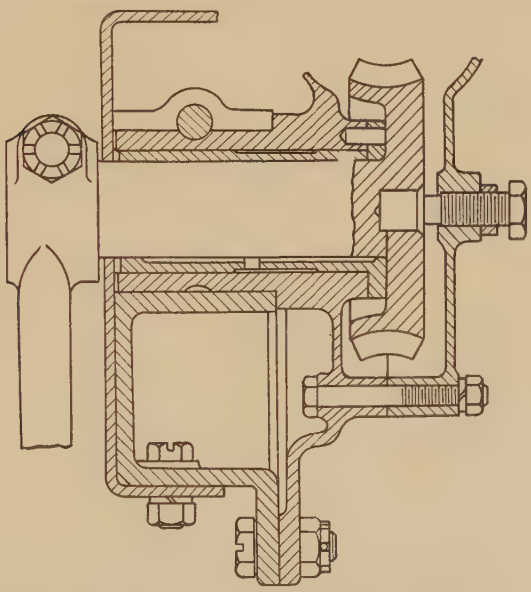


FIG. 15.—TRUNNION-MOUNTING OF GEAR FOR ADJUSTMENT OF STEERING-COLUMN RAKE.

lugs by which they can be bolted to the frame side member. In addition, the column is braced by means of a bracket at about the middle of its length, which is secured to the cowl-board. In many cases the steering-gear shaft passes through a hole through the web of the frame side rail. In Europe a good deal of importance is attached to means for adjusting the rake of the steering column, so it can be made to suit the individual owner. This is usually accomplished as illustrated in Fig. 15. The bearing hub on the housing is turned down on the outside to form a trunnion, and is supported in

a trunnion block secured to the frame side rail, the block usually being split and provided with clamping bolts. A sector-shaped bracket is mounted on the dash, with a number of bolt holes in it, and the wheel end of the column is raised or lowered to bring a bolt hole in a lug on the column into register with one or the other hole in the sector, after which a bolt is passed through and the column clamped tight both at the lower end and at the dash.

If the drag link is outside the frame, it limits the angle through which the wheel on that side can be deflected from the straight-ahead position. In order to minimize interference between the drag link and front wheel, the steering-gear arm sometimes has its lower end bent inward, or is provided with a ball stud extending from it toward the center of the chassis.

Drag Link and Connectors—In most cases the drag link

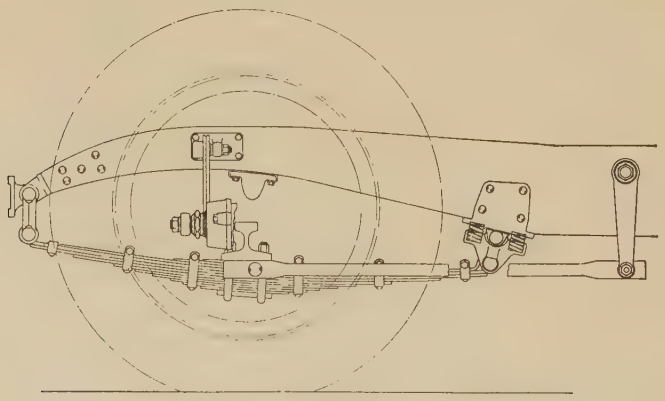


FIG. 16.—CONNECTION OF SPRING TO FRAME FOR MINIMUM EFFECT OF SPRING-ACTION ON STEERING.

extends substantially in a fore-and-aft direction, though where the driver's seat is not far behind the front axle, it is more convenient to have it extend crosswise of the chassis and connect to the steering knuckle on the opposite side of the chassis from that on which the steering column is located. In every case the object should be to so arrange matters that the play of the front springs will have the least effect on the steering mechanism. There will always be a slight effect of this kind; that is, with the steering gear set for straight-ahead motion, for instance, a compression or rebound of the front spring on the side of the car where the drag link is located,

will move the front wheels to one side or the other of the straight-ahead position. Various means are available for reducing this effect. Making the drag link long as compared with the play of the springs will help. With a fore-and-aft drag link back of the front axle and semi-elliptic front springs, the best arrangement is to pin-joint the rear and shackle the front end of the spring, as illustrated in Fig. 16. Then the front axle will describe an arc around the axis of the rear spring bolt, while the forward end of the drag link will describe an arc around the center of the steering-gear arm ball, and there will be very little deviation between these arcs, which may be made to coincide when the springs are in their normal position. The drag link should be parallel with the rear part of the front spring when both are in the normal position. This arrangement has been used by De Dion-Bouton and several other makers.

Not quite such good results are obtainable if the front end of the front spring is pin-jointed to the frame and the

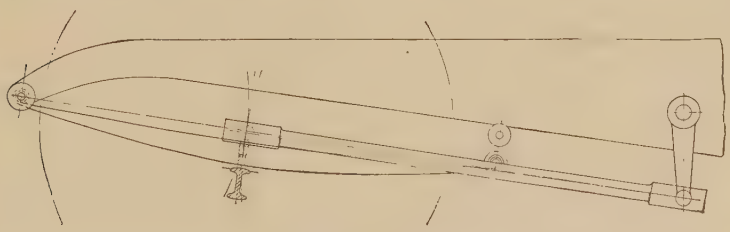


FIG. 17.—BEST LAY-OUT OF FRONT SPRING AND DRAG LINK WHEN REAR END OF SPRING IS SHACKLED.

rear end shackled. In that case, as illustrated in Fig. 17, as the front spring compresses and rebounds, arcs are described by the front axle and the front end of the drag link which are tangent to each other but whose centers, instead of being on the same side, are on opposite sides of the point of tangency, for which reason there is greater deviation between them. The deviation will be a minimum when the drag link is in line with the forward section of the spring when the spring is under normal static load, as shown in Fig. 17. If the drag link extends across the frame, it preferably should be horizontal when the chassis is normally loaded.

The drag link is usually made of tubular form, and its inside and outside diameters are made the same as those of the tie-rod. If it is very much shorter than the tie-rod, its

section can be made smaller to advantage, and can be calculated by making use of Rankine's formula for columns, assuming the same maximum load as in the case of the tie-rod and determining a section which has the same factor of safety with the smaller length. In general, however, very little weight could be saved, and a little higher factor of safety in this important part is not undesirable. Moreover, there is a slight manufacturing advantage in having both the tie-rod and the drag link of the same stock, since both, as a rule, are fitted with connectors of the same type.

In cars of the better grade it is customary to introduce cushion springs in the drag-link joints to relieve shocks. A length of tubing of enlarged diameter is screwed over the end of the tubing forming the main part of the drag link, or is pinned or welded to it. As shown in Fig. 18, inserted into this connector housing are the following parts in the order named: A coiled spring, a spring block with a ball socket, the ball end of the steering arm, another spring block, another coiled spring and a screw plug secured by a cutter pin.

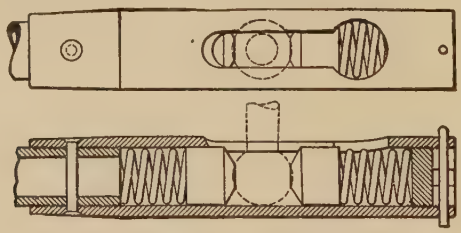


FIG. 18.—CUSHIONED DRAG-LINK CONNECTOR.

In the particular design shown, the ball is passed through a hole far enough to the end of the connector housing so that the ball can never get opposite it after all of the parts are in place. Sometimes the slot through which the steering arm passes extends entirely to the end of the housing, in which case the end of the housing is closed by a screw cap instead of by a plug. The spring-cushioned joint is generally placed at the steering-gear end of the drag link, the joint at the opposite end being made similar but without the springs. Some means are generally provided for closing the opening around the end of the steering-gear arm, so that lubricant cannot get out, nor dust in. Lubricating fittings are provided on both joints.

Steering Arm—Steering arms for passenger cars are made from 5 to 9 in. in length, the proper normal inclination of the drag link being one of the factors to consider when

determining this length. In heavy commercial vehicles the length can be made 9 to 10 in. The steering-gear arm is proportioned as indicated in the dimensional drawing Fig. 19. It is secured to the steering-gear shaft by means of a serrated joint, which has a slight taper, the arm being forced onto the shaft by a castellated nut with a washer under it. The steering arm can be made with an integral ball for the ball-and-socket joint at its lower end, but the preferred practice is to use a ball stud which is secured in the lower boss of the arm by means of a taper seat and nut. Ball studs have been standardized and the standard dimensions are given in the Appendix.

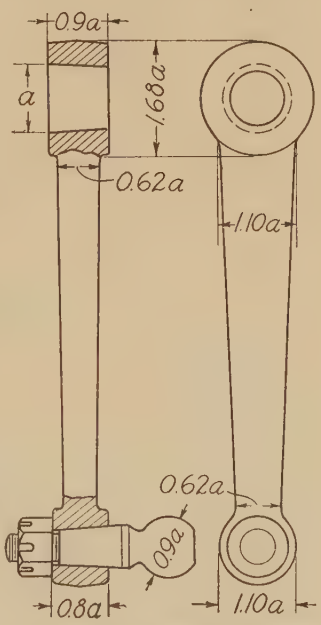


FIG. 19.—DIMENSION DRAWING OF STEERING ARM.

Steering Wheel — Steering hand wheels are made 17 in. in diameter for the smaller cars, 18 in. for large passenger cars and the smaller trucks and buses, and 20-22 in. for the heavier trucks and buses. The wheel spider, which generally has four spokes, is made of malleable iron or aluminum for commercial vehicles and of aluminum, hardwood or bronze for passenger vehicles. The hub is bored out to fit the steering post, and cut with a keyway. It is counterbored from the lower end to be a loose fit over the steering column jacket. Steering wheel hubs have been standardized by the S.A.E. Spokes are mostly made of oval section, tapering from the hub toward the rim. For a light car the arm section of an aluminum spider may taper from $1\frac{1}{4}$ by $\frac{5}{16}$ in. near the hub to $\frac{7}{8}$ by $\frac{3}{16}$ in. near the rim, while for a heavy passenger car or bus it may taper from $1\frac{1}{2}$ by $\frac{1}{2}$ near the hub to 1 by $\frac{1}{4}$ near the rim. The spokes join both the hub and the rim with liberal fillets. Sometimes sections of T or channel form are used to get greater stiffness in the axial direction for a certain amount of material in the wheel, but a full section of rather small axial depth seems to be preferred, because it gives a certain amount of elasticity and reduces fatigue in the driver's arms.

When the steering wheel has wooden spokes, these combine in a ring or hub at the center, of the same depth as the part of the spoke near it. A metal hub in two parts—lower and upper—is fastened to the wood hub by being clamped over it by means of machine screws, the torsional moment being transferred from the wooden to the metal members by steel dowels which pass entirely through holes in the wooden hub and into holes in the metal hubs.

Steering-wheel rims are made of hard wood or hard rubber. For the more expensive cars, rims of mahogany and walnut are used. Wood rims are generally made of oval section, 1 by 1⅜ and 1⅛ by 1½ in. There are two general designs of wood-rim wheels. In one the spokes of the wheel are secured to the rim by means of wood screws, while in the other the spider is cast with an integral metal rim which is let into a part of the wood rim.

In some cases the rims are made of solid wood, the whole

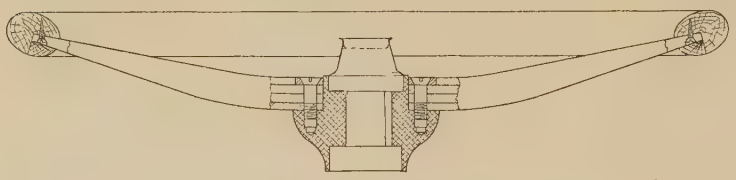


Fig. 20.—Steering Wheel with Wood Spider and Wood Rim.

wheel being made either of a single piece, steamed and bent to shape, or else made of a number of arcs. First a solid piece of wood of square section and the requisite length is sawed out, and its ends are cut with wedge-shaped teeth about ¼ inch wide and 1 to 1¼ inches deep. It is then turned down to a circular or oval section, steamed and bent to a circle of the right size to bring the pointed tenons together, when they are glued and firmly pressed together in a special clamp in which the rim is left until dry. Specially flexible woods are used for this construction, as most woods do not allow of bending to such a small radius. Instead of making the entire rim in one piece, it may be made in halves, with joints of the type above described. A quarter-inch dowel pin through the center of the joint makes it more secure.

Another method of making the rims consists in building them up of segments. As a rule, three layers of segments of four or six to the circle are used, six being preferable, as there is not so much end grain where the segments are joined together. After the segments are sawed out, they are glued

up three deep into rings, and are then turned up in a lathe. They are secured in the lathe either by being glued to a wooden face plate with a piece of newspaper between, or else by means of a number of screws. If the spider is cast with an integral ring, the final turning up has to be done with the spider in place. Figure 20 illustrates a steering wheel with wood rim and wood spider. A wheel with wood rim and metal spider is shown in Fig. 21.

Rubber Rims—Steering-wheel rims of hard rubber have the advantage that forms of special, more or less irregular section to give a better hand-grip can be obtained at practically no extra expense, as the rims are formed by molding. Wooden rims, instead of being made perfectly round, are sometimes made in the shape of a polygon with rounded corners, and both wooden and molded rubber rims are frequently made with either projections or a scalloping effect either on the bottom or on the inside, to give a better hand-

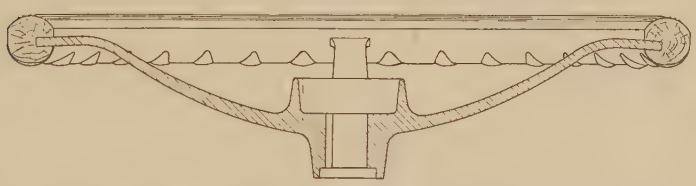


FIG. 21.—STEERING WHEEL WITH METAL SPIDER AND WOOD RIM.

grip. The upper surface is sometimes provided with light corrugations concentric with the rim.

Safety in Steering Gears—Every part of the steering mechanism must be designed with an eye to a high degree of safety. The major parts, such as the steering knuckles and arms, are made of high-grade alloy steels, carefully heat-treated. All bolted connections must be positively locked, preferably by both lock washers and cotter pins. The tie-rod and drag-link tubes must be of such wall thickness that if threads are cut on them for the connectors, these will not reduce their strength inordinately. Welded and pinned (or riveted) joints between these tubes and their connectors are preferred by many engineers. If a threaded joint is to be used, it is advisable to counter-bore the threaded hole in the connector for some distance to a snug fit over the unthreaded portion of the tube, slotting the hub of the connector lengthwise and clamping it down over the tube with a substantial clamp bolt. The end of the connector hub should be tapered

down to an edge, in order to prevent concentration of stress where the tube leaves the connector. If a portion of the tube outside the connector hub were threaded, that would be a particularly weak section.

Turning Radius—The turning radius of a motor vehicle may be measured in different ways, viz., to the center of the track of the outside front wheel, to the outside of the wheel or rim of the outside front wheel, and, finally, to the most projecting part of the vehicle, which is usually the tip of the outside front fender. In order to avoid confusion due to inexactness of nomenclature, the Society of Automotive Engineers adopted the following definition: "The turning radius of an automotive vehicle is the radius of the arc described by the center of the track made by the outside front wheel of the vehicle when making its shortest turn."

This turning radius depends upon the wheelbase w (Fig. 22), the distance b between knuckle pivot axes, the maximum angle θ through which the inside front wheel can be deflected from the straight-ahead position, and the distance d from the intersection of the front-wheel axis with the knuckle-pivot axis to the center plane of the front wheel. Referring to Fig. 22, by one of the rules for the solution of the oblique triangle,

$$c = \sqrt{a^2 + b^2 - 2ab \cos (180^\circ - \theta)}$$

but

$$\cos (180^\circ - \theta) = - \cos \theta,$$

hence

$$c = \sqrt{a^2 + b^2 + 2ab \cos \theta}.$$

If now we substitute the value of a in terms of the wheelbase and angle of lock ($a = w/\sin \theta$), we have an expression for c entirely in known terms:

$$c = \sqrt{\left(\frac{w}{\sin \theta}\right)^2 + b^2 + \frac{2wb}{\tan \theta}}.$$

The turning radius is equal to $c + d$, and hence is

$$R = \sqrt{\left(\frac{w}{\sin \theta}\right)^2 + b^2 + \frac{2wb}{\tan \theta}} + d \text{ in.}$$

However, the turning radius is most conveniently found graphically, by laying off the plan of the chassis to scale, with the front wheels deflected to the limit, and measuring the distance from the point where the axes of the front-wheel spindles produced intersect the axis of the rear wheels, to the center of the outside front-wheel track.

Steering Radii of Trailers—Each wheel of a four-wheeled vehicle, as a rule, describes a separate track, but it is usually most important to know the radius of the track described by the outside front wheel, as it is the largest. However, under certain conditions it is desirable to know also the radius of the track described by the inside rear wheel, as the distance apart of these two tracks, which is equal to the difference between their radii, determines whether a vehicle or tractor-trailer combination can negotiate the turn at intersecting streets of given widths. The radius of the track described by the inside rear wheel is also best determined graphically. For the case of a four-wheeled vehicle it can be measured off

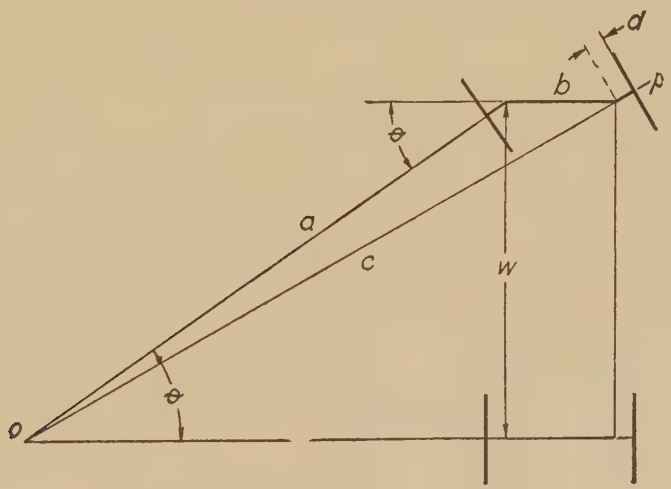


FIG. 22.—TURNING-RADIUS DIAGRAM.

directly from a diagram similar to that of Fig. 22. For a tractor and trailer combination the diagram takes the form shown in Fig. 23. That portion pertaining to the tractor is identical with the diagram Fig. 22. Let L be the distance from the fifth wheel axis of the trailer to the center of its rear axle. The distance R, from the turning axis to the center of the rear axle of the tractor, can be determined by measurement on a diagram similar to Fig. 22. Now, the axis of the rear axle of the trailer evidently must intersect the turning axis, and the angle ϕ between the axis of the two rear axles therefore must be such that $L/R = \sin \phi$. After this angle has been determined, the plan of the trailer can be laid off,

and the distances from the turning axis to the center of the trailer inner wheel can then be measured off. The distance R from the turning axis to the center of the trailer rear axle is equal to

$$\sqrt{R^2 - L^2}$$

and the distance to the center of the rear inner track evidently is less by half the tread.

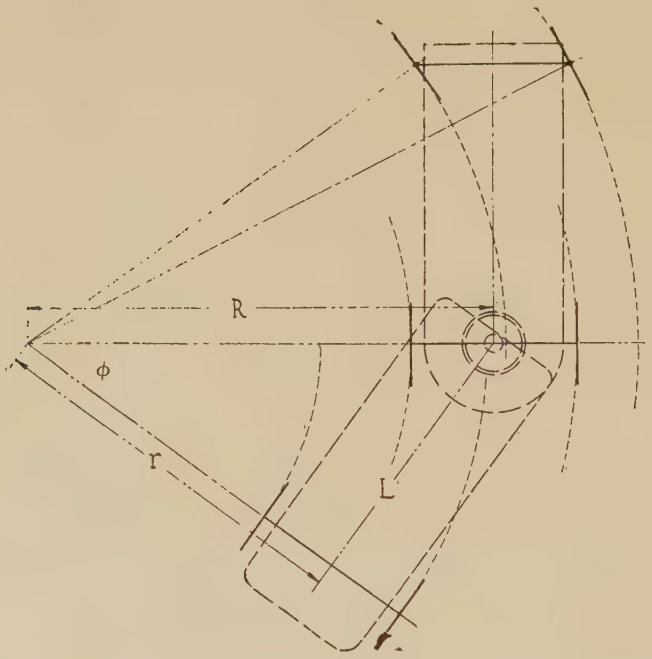


FIG. 23.—TURNING-RADIUS DIAGRAM OF TRACTOR AND TRAILER.

Where a vehicle has to be designed so as to be capable of turning in very close quarters, it is made to steer by both the front and the rear wheels. If both wheels on the same side of the vehicle are deflected equally from the straight-ahead position, the front and rear wheels will "track."

Steering of Six-Wheelers—Since perfect steering requires that the axes of all of the wheels should always pass through a single vertical line, it is obvious that the six-wheeler cannot have perfect steering; for the axes of the two pairs of rear wheels are parallel. Figure 24 shows a steering diagram for

such a vehicle. There must, of necessity, be a small amount of slippage of the rear wheels on the ground when describing curves, but it has been shown that if the two driving axles are fairly close together, this slippage is less than that with large dual rear tires which, although turning at the same angular speed, must describe circles of different diameters while describing curves.

Shimmy and Wheel Wobble—With high-speed vehicles fitted with low-pressure pneumatic tires, a good deal of trouble has been experienced from wheel wobble and shimmy. Wheel wobble consists of periodic vibration of the front axle assembly around a horizontal longitudinal axis through its center. The front axle is mounted between two "springs,"

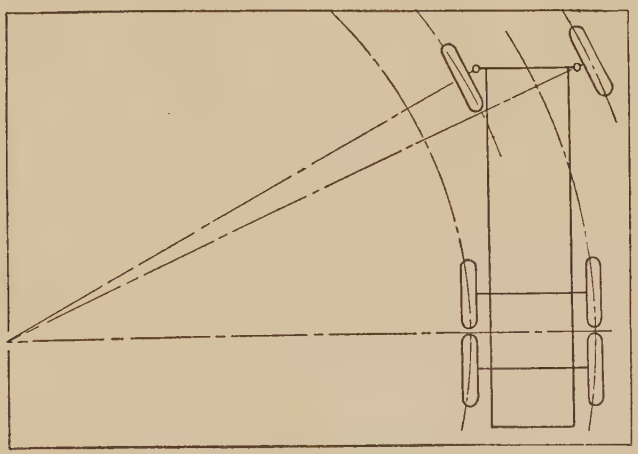


FIG. 24.—TURNING-RADIUS DIAGRAM OF SIX-WHEELER.

the body spring and the tire, and when there is wheel wobble, the body spring on one side and the tire on the other compress simultaneously, while the other spring and tire rebound. Such vibration can become troublesome only if there is synchronism between a periodic exciting force and the natural period of the vibrating body. The amplitude which may be attained by the vibration depends on the exciting force on the one hand, and on the damping forces on the other. The natural period of vibration of the vibrating body, the front-axle assembly, is directly proportional to the moment of inertia of the assembly around the axis around which vibration takes place, and inversely proportional to the restoring forces, the spring

forces of the tires and chassis springs, which may be expressed as so many pounds per inch of compression.

In order to produce wheel wobble, the rate of impulses must be equal to or a simple fraction of the natural rate of vibration of the front assembly. The rate of impulses, however, for a given road conformation is directly proportional to the speed of the vehicle. If the vehicle has a low-speed limit and the rate of natural vibration of the front assembly is high, owing to low weight, and the use of high-pressure tires and stiff chassis springs, the road impulses never come fast enough to cause synchronous vibration. That was the case with the great majority of cars at least up to the advent of the low-pressure tire and the front-wheel brake. The adoption of both of these features resulted in an increase in car speeds, thus increasing the rate of impulses received by the front assembly; and it at the same time lowered the natural rate of vibration of the front assembly, because the addition of front-wheel brakes meant a greatly increased moment of inertia, while the adoption of low-pressure tires meant a reduced restoring force, both of which lower the natural rate of vibration. As a result, with modern passenger cars, synchronism between vibration-inducing impulses and the natural period of vibration occurs quite frequently.

The above explains the phenomenon of wheel wobble or vibration in a vertical plane. The other phenomenon referred to in the foregoing, wheel shimmy, consists of vibration (a flapping) of the front wheels around the knuckle pivot axis. Shimmy is closely related to wheel wobble. When the car is in motion, the front wheels possess the properties of spinning tops or gyroscopes. When the axis of a gyroscope is moved angularly in a given plane, a force is set up which tends to move the axis in a plane at right angles thereto. Now, whenever there is wheel wobble, the axes of the front wheels are moved in a vertical plane, and the logical consequence is that forces on the front wheels are set up which tend to cause them to move in a horizontal plane. Since the movement in the vertical plane alternates in direction, the forces causing movement in a horizontal plane also alternate in direction, and thus a flapping motion around the knuckle pin axis is produced which is known as wheel shimmy.

There are, however, other reasons which will serve to explain the interrelation between wheel wobble and shimmy. Wheel wobble involves alternate compression and rebound of the chassis springs. Whenever the chassis spring on the side where the steering gear is located compresses, if the steering gear is very rigid and irreversible, the front wheels are com-

pelled to go through a slight angular motion around the knuckle pin axis, owing to geometrical imperfections of the steering linkage. For instance, if the forward end of the front spring is pivoted to the chassis frame, when the spring compresses, the axle will describe an arc around the front spring front eye axis as a center, while the drag link will describe an arc around the center of the ball on the steering-gear arm as an axis. These two arcs will deviate from each other as the spring compresses and rebounds, and this deviation must be compensated for by angular motion of the front wheels around the knuckle pin axis.

These two relationships between the causes of wheel wobble and shimmy probably explain why there are two types of shimmy, referred to, respectively, as pseudo-shimmy and true shimmy, or low-speed shimmy and high-speed shimmy. On the other hand, it may be that low-speed shimmy is the result of road impulses constituting a lower harmonic of the natural rate of vibration of the front assembly. That is, instead of the front end receiving an impulse at every cycle of its vibration, it may receive one only every other cycle.

There are evidently four factors affecting wheel wobble and shimmy, viz., (1) the rate of impulses due to road shock, which depends upon road conditions and car speed and which the designer cannot alter; (2) the moment of inertia of the front system around a horizontal axis through its center; (2) the restoring force, which depends upon the "stiffness" of the tires and springs, and (4) the damping action. If the moment of inertia of the front system can be lowered by reducing the weights of the wheels and brakes, and if the restoring forces can be increased by using higher inflation pressures and stiffer springs, it may be possible to raise the point of synchronism to such a high car speed that it is outside the ordinary driving range. These factors, however, cannot be changed much, as a rule, without undesirable consequences, and the remedy that is most available is an increase in the damping forces.

Sensaud de Lavaud, a French engineer, has suggested the use of independent springing for each front wheel as a preventative against shimmy. Charles Waseige, another French engineer, recommends the use of indendent connections from the steering gear direct to each steering knuckle. Evidently, if each front wheel is indendently sprung and also indendently connected to the steering gear, there is no rigid connection between the two, and they no longer form a single vibrating system, as they do in the conventional vehicle.

The method of combating wheel wobble and shimmy in

most extensive use consists in providing extra damping means. Friction at the knuckle pivots and in the chassis springs provides a damping action, which, however, is not sufficient to prevent the phenomena when the exciting forces are very strong. In some instances friction dampers have been inserted directly on the knuckles or between the tie-rod and the axle center, but such friction dampers, while damping shimmy, also make the steering gear harder to operate.

A more practicable scheme appears to be to so mount the shock absorbers with which cars are usually fitted, that they attach to the axle as close to the wheels as possible (Fig. 25), which has been suggested by M. Causan, a French engineer. Shock absorbers of the mechanical type consist of two sets of friction plates, each provided with a radial arm, one arm being connected to the axle and the other to the car frame. Usually these shock absorbers are arranged in a fore-and-aft plane close to the frame side member, but by placing them in a transverse plane, with the arms extending toward the

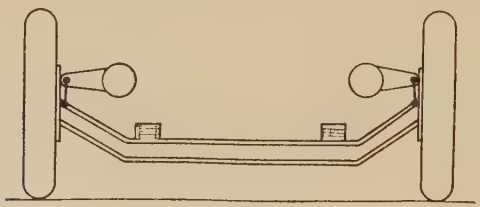


Fig. 25.—Mounting of Shock Absorbers for Best "Shimmy"-Damping Effect.

wheels, their damping action on wheel wobble can be materially increased.

F. W. Lanchester, an English engineer, has invented an hydraulic damping device for wheel shimmy (Fig. 26). This consists of a semi-rotary piston working in a segmental chamber completely filled with oil. Valves are provided to by-pass the oil from one side to the other when the steering gear is worked in the regular manner by the hand wheel, but any reaction from the road is checked by the hydraulic system, which, in itself permits of only a small degree of leakage. This leakage is essential in order that the steering gear should not be rendered irreversible. It allows the front wheels to "straighten out" after turning a corner, but it does not permit of any rapid movement, such as gyroscopic kick, due to road irregularities. When using a shimmy damper of this type, there must be no spring connectors in the steering linkage,

as they defeat the object of the damper. It is still possible, however, to place a flexible member between the road wheel and the steering wheel. The jacket over the steering shaft may be terminated at the bracket which supports the steering column at the cowlboard. Then the lower end of the steering shaft is exposed, and it may be cut in two and have a flexible joint of the fabric type inserted between the two parts. This method, however, is readily applicable only with steering gears having control members outside the column.

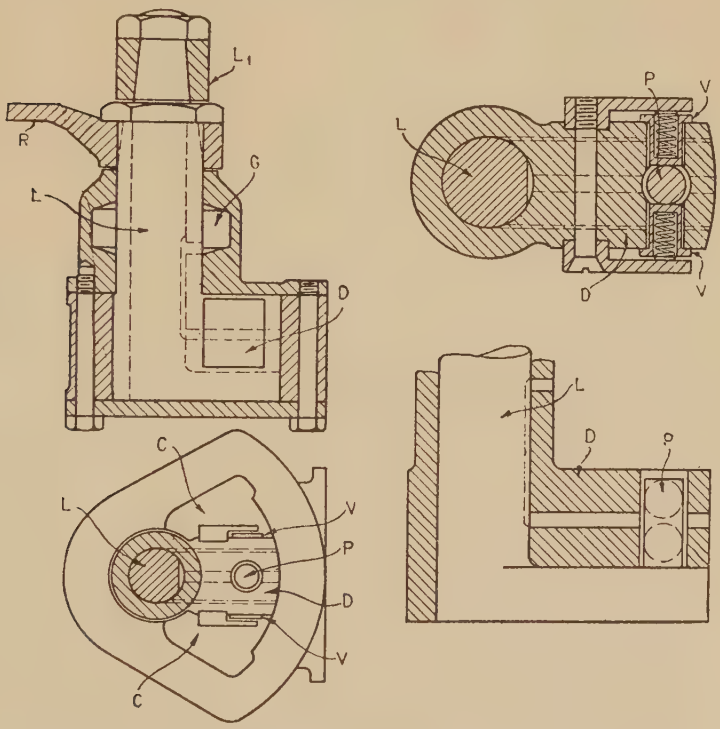


FIG. 26.—SECTIONAL VIEWS OF THE LANCHESTER HYDRAULIC "SHIMMY"-DAMPING DEVICE.

CHAPTER XIV

Control

The devices by means of which the operation of the engine is controlled may include any or all of the following: A throttle lever, a throttle pedal (accelerator pedal), a spark-timing lever, a choke button or lever, a mixture-control lever, a vaporizer-temperature-control lever. Of these the throttle lever is used most frequently, while the spark lever is also used a good deal, unless the spark is automatically controlled. For this reason the spark and throttle levers are generally mounted on the steering column, where they are most handy.

There are two methods of arranging these more important levers and their connections to the parts which they control. The simplest consists in placing the levers on the steering column under the wheel, with their shafts extending parallel to the column and supported in bearings mounted on the column. This plan, which permits of the use of a solid steering-wheel shaft, is used on motor trucks and some of the lower-priced passenger cars. These controls, however, are much handier to operate if the levers are on top of the steering wheel, with their connections extending through the steering-wheel shaft.

In passenger cars of the better grades it is now the common practice to have the spark and throttle levers, a horn button and a headlight control button or lever all mounted concentrically on top of the steering wheel. The headlight control switch generally is mounted at the bottom of the steering column and is operated through a tube extending through the column. Tubes are required also for the throttle and spark levers, and there must be one stationary tube to support a sector or friction disk on which the various levers can be locked by friction or by means of a ratchet in any position within their range. All of these tubes are arranged concentrically, and through the innermost of them extends an electric cable carrying current to the horn button.

Ratchet Lock—Occasionally the ratchet-type of lock is still used for the control levers, though it has been largely superseded by the friction lock. A design of control mechanism with ratchet lock is illustrated in Fig. 1. In this particular case there is no horn button at the center of the steering wheel, and one of the levers is connected by pinning to a solid shaft, while the other is connected to a tube surrounding the shaft, by means of a clamp bolt. Surrounding this there is another stationary tube which supports a metal spider to which a steel sector is secured with machine screws. This sector is formed with teeth on both its inner and outer edges. Each lever is provided with a spring-pressed pawl which

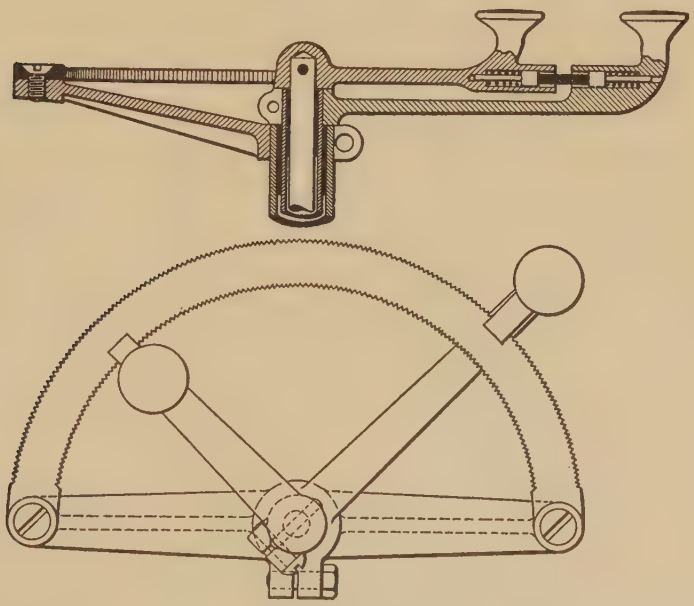


FIG. 1.—RATCHET CONTROL.

engages with the teeth on the sector. The teeth on the sector form angles of about 90°, so that if pressure is applied to the lever, the pawl will slide freely over the notched sector.

Friction Lock—Locking of the control wheels by friction is generally preferred, not only because it does away with the grating sound of the pawl sliding over the teeth of the sector, but also because it lends itself to a more compact and a neater design. The friction lock can be located either at the top of the steering column or at the bottom of it.

A friction-lock control mechanism of the Gemmer Mfg. Co. is shown in Fig. 2. It will be seen that there are four concentric tubes. The whole assembly is passed through the tubular steering-wheel shaft, and to prevent rattling of the control tubes against the steering-wheel shaft, the latter are surrounded by anti-rattlers of spring brass, of which one is shown sectioned in black at the bottom of the figure. The outermost tube is non-rotatable and has a U-shaped steel stamping A fastened to it. The method of fastening it is worth noting. The end of the tube is swaged into a steel bushing which is conical on the outside over part of its length, and is threaded over the remainder. The U-shaped stamping is formed with a tapered opening in it and is fastened to the conical part of the bushing by means of a nut.

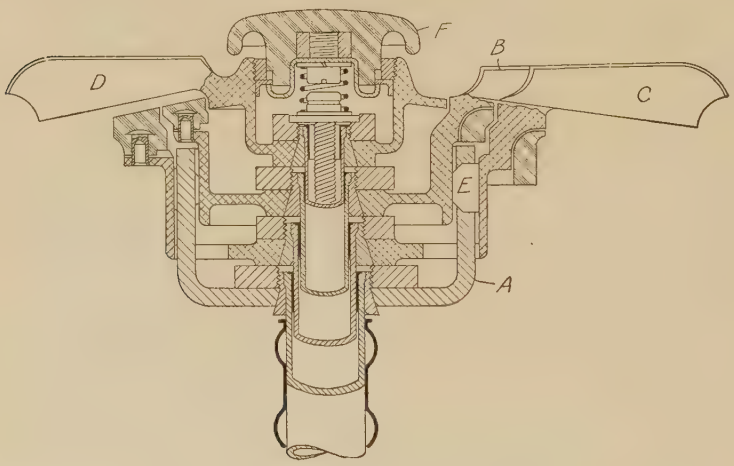


Fig. 2.—Friction-Lock Control.

This same method is used for securing the three levers, B, C and D to their respective tubes. The hub portions of the various levers are in the form of cups which are located one within the other. Most of this assembly is located in a depression at the center of the steering wheel. The U-shaped stamping A serves as anchorage for the spring plate E which locks the levers B and C in position by its friction against their hubs. These levers B and C carry rings of bakelite on top, which serve for purposes of ornamentation only. Lever D, which controls the headlights, requires no special lock, as the light-control switch at the bottom of the steering column

serves as such. In the center of the entire assembly is the horn button F, which is made of insulating material and has a metallic insert on its lower side. Directly below this insert there is another button to which the upper end of the cable coming up the steering column is connected. The horn button is normally held up against a stop by a coiled spring, and when it is pressed down the two metal buttons come in contact and establish electrical connection between the horn cable and the whole control assembly, that is, with "ground."

A somewhat different design of friction control (Reo) is shown in Fig. 3. In this case, too, there are four concentric

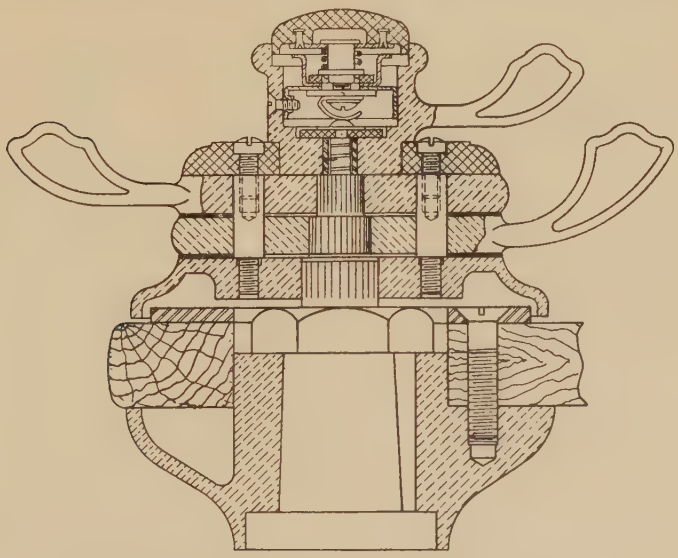


FIG. 3.—FRICTION-LOCK OF "PANCAKE" TYPE.

tubes, with a cable passing through the smallest of them. The outermost tube is stationary (being clamped in the hub at the bottom of the steering-gear housing). At the top it carries a circular base, into which, at diametrically opposite points, are screwed two studs. Directly above this base, and separated from it by a layer of friction material, is the disc-shaped hub of one of the control levers. The disc is secured to its tube by an expanding process and is provided with two arc-shaped openings through it, so the studs in the base below it can pass through it without preventing it from being moved angularly through a certain range. Next above this

there is another disc-shaped hub of a control lever, separated from the hub below it by another disc of friction material, and then follows a stationary disc which is securely fastened to the base plate by the studs referred to, and machine screws screwed into them from the top. The hub or center of the light-control lever, which is the uppermost one, has a support in this top plate, and it in turn supports the horn button.

Another scheme for placing the control levers in driving connection with the control tubes consists in fastening flat spiders to the control tube by a riveting or swaging process and forming the levers with hubs with slots on the inside, into which the arms of the spider engage when the control levers are assembled.

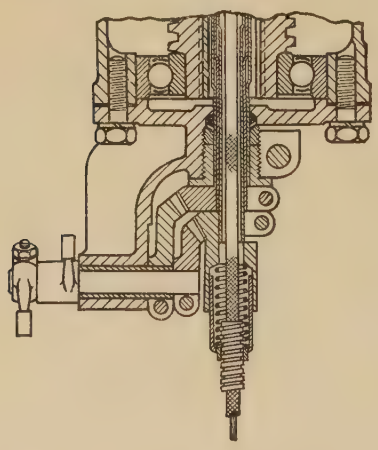


FIG. 4.—BEVEL-TYPE CONTROL REDUCING GEAR.

Mechanism at Bottom of Column—From the bottom ends of the control tubes the motion may be transmitted in different ways. The simplest plan is to fasten a lever directly to each tube and run a linkage from it to the part to be controlled. The usual method of fastening the lever in that case is by a clamp screw, and the tube may be knurled or milled at the point, to ensure a more secure hold. However, the control levers on top of the steering wheel usually have a range of motion of about 150° each, and if the lower lever arm has a link connected to it, the motion given to this link varies greatly over the range of motion of the control lever. To obtain a more nearly uniform transmission of the motion, a reducing gear is often employed, and one of the most popular types of reducing gear is a bevel pinion and sector as illustrated in Fig. 4. The two sectors are mounted on concentric shafts which carry lever arms beyond their common bearing.

There are various other methods of making connections from the lower ends of the control tubes to the parts to be controlled. Sometimes a cam is employed, and this may be either a radial cam or a face cam. The cam acts on a lever

which is held in contact with it by a spring. It is also possible to use a drum with a cam groove in it, in which case the spring is not needed. Both direct connection from a lever and reduction by means of a cam are illustrated in Fig. 5.

Control Levers on Steering Post—Figure 6 illustrates a construction in which the control levers are mounted on the steering post underneath the wheel. This design is used more particularly on commercial cars, although many of the smaller passenger cars manufactured in Europe have a similar arrangement. The shafts for the spark and throttle are shown arranged concentrically and are supported in bearings secured to the steering column. A sector also is cast integral with the supporting bearing at the top, and the control levers are pressed into contact with it by means of a coiled spring

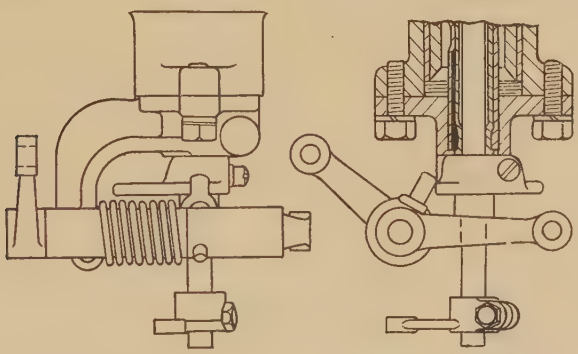


FIG. 5.—CAM-TYPE CONTROL REDUCING GEAR.

surrounding the lower part of the central control shaft, the spring pressing the tubular shaft upward and the solid shaft downward. Lever arms are secured to the lower ends of these shafts, and connections to the throttle and timer are made by links direct. The sector over which the levers move extends over an angle of about 90°, and is placed in such a relation to the steering column as to make the linkage the simplest. Instead of having the two control shafts concentric, it is not uncommon abroad to have them located on opposite sides of the column.

Bowden Wire Mechanism—It is customary to control the carburetor choke valve (which serves to facilitate starting of the engine) from the dash by means of a Bowden wire mechanism, or a mechanism similar thereto. The original Bowden wire mechanism consisted of an outer member com-

posed of a closely-coiled and practically incompressible wire, and an inner member in the form of a practically inextensible cable threaded through the outer member. The principle of the mechanism is illustrated in Fig. 7, where the control end is shown mounted on the steering column, although at present it is commonly mounted on the instrument board. A is the actuating lever; B, the operated lever; C, the inner member; D, the outer member; E, an adjustable stop and F, a lock nut, while G, G are abutments or brackets.

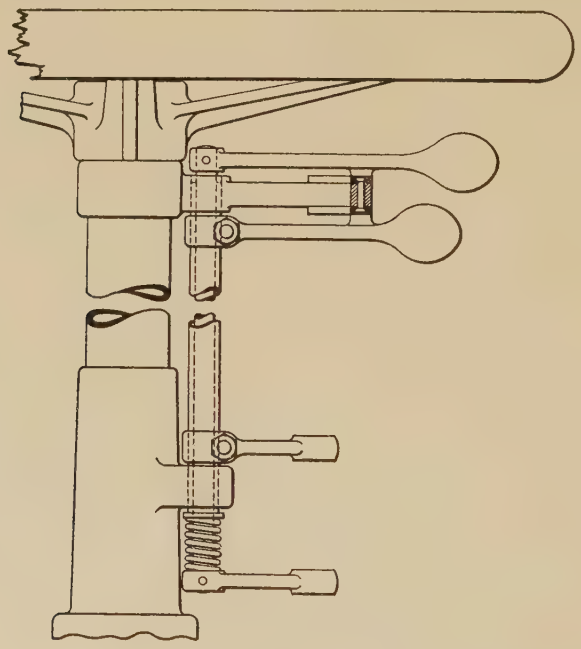


Fig. 6.—Control Levers on Steering Column.

It is obvious that, since the outer member is incompressible and the inner member inextensible, if lever A is moved around its fulcrum, the end of lever B will be moved with relation to the abutment G.

Control Joints—In the connecting linkage, if two lever arms to be connected swing in the same plane, a forked connector is employed, and a standard for such connector yokes and eyes has been worked out by the Society of Automotive Engineers. If the two arms do not swing in the same

plane, as is often the case, a ball-and-socket type of joint is used. The links of the control mechanism generally are made of cold rolled steel, 3/16 or 1/4 in. in diameter, and one part of the joint is screwed over the end of the rod. In cars sold at a very low price the end of the rod is sometimes bent at right angles and passed through a hole, of slightly greater diameter, in the end of the arm, a split pin being passed through the end of the rod. This type of joint rattles more or less ,and is not very satisfactory. Figure 8 shows three types of ball-and-socket joints for carburetor and spark connections. The one shown at A consists of a brass socket and a steel ball, the brass socket being bored out and having the

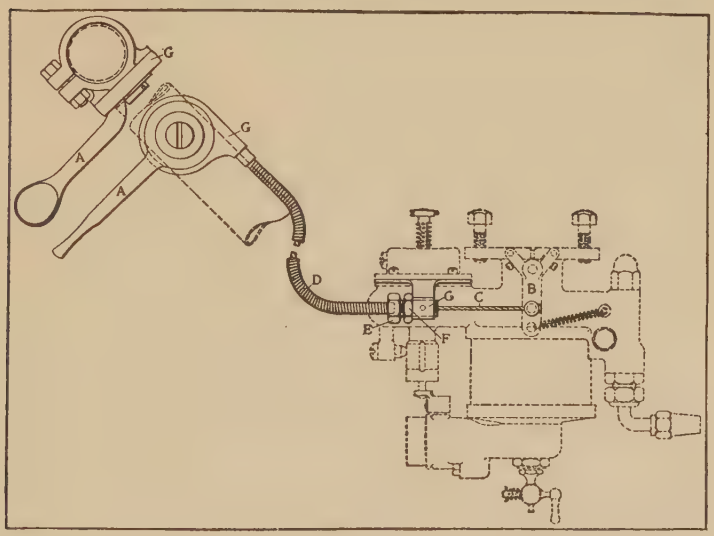


FIG. 7.—BOWDEN WIRE MECHANISM.

edges spun in after the ball is in place. A well-designed type of joint is shown at B. This resembles a steering drag-link joint, except that only one spring is used whose object is to firmly press the socket blocks against the ball. No play can develop in a joint of this type, and therefore it remains free from rattle. The joint shown at C is similar except that the spring is missing and D shows the simple joint above referred to.

Usually at least one of the devices that must be connected to the control levers is located on the opposite side of the

chassis from the steering column, and the connection to it must then pass either around or through the engine. Passing a shaft transversely through the crankcase above one of the crankshaft bearings eliminates unnecessary linkage. Great care is latterly exercised by designers to render the control linkage as simple and unobtrusive as possible as that not only improves the appearance of the chassis but often makes vital parts more accessible.

Accelerator Pedal—On the majority of cars both hand control and foot control for the throttle are provided. The lever or button by means of which the throttle is opened by the foot is generally known as the accelerator or accelerator pedal. Quite frequently this is in the form of a plunger extending through a guide in the toe-board, as shown in Fig.

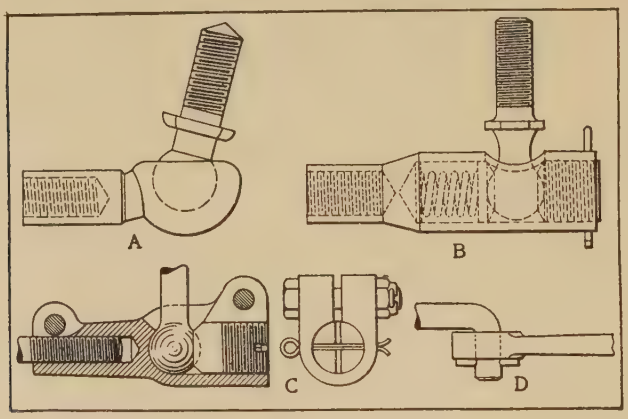


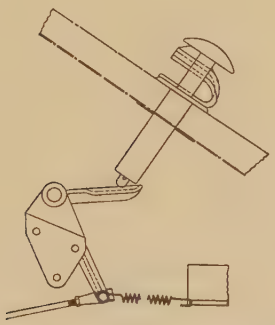
Fig. 8.—Control Joints.

9, the lever or bell crank being below the toe-board. When this type of accelerator is used it is now common to provide a foot rest on the toe-board adjacent to it. The driver's right foot (with which the accelerator is always operated) then covers both the foot rest and the accelerator button, and the accelerator is depressed by inclining the foot sideways. Figure 10 shows a design of accelerator comprising a lever fulcrumed below the toe-board and having a wedge-shaped spur which projects through a slot in the toe-board. This is operated by sliding the foot forward over the toe-board. The accelerator is always returned to the position corresponding to closed throttle by a spring.

When the throttle is opened by means of the accelerator

pedal, the throttle lever on the steering wheel must not be affected. This calls for a sliding connection in the linkage from the throttle lever to the throttle, with enough range of motion to permit the throttle to be moved from the closed to the fully open position. It is possible also to so arrange the linkage that there will be no motion of the accelerator from the "fully-back" position when the throttle is opened by hand, but there is no particular advantage in this. The sliding connection may take the form of a pin extending through a slot, a rod with collars on it extending through a sleeve, or a lug on one lever which takes along another, adjacent lever.

Motor trucks generally have their engines fitted with governors, and require a special control mechanism. Two sepa-



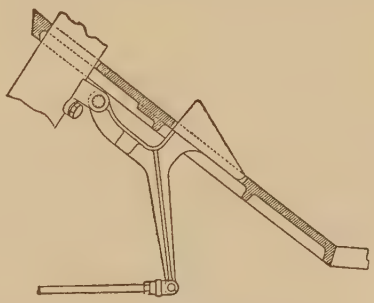


FIG. 9 (LEFT).—PLUNGER-TYPE OF ACCELERATOR.
FIG. 10 (RIGHT).—WEDGE-TYPE OF ACCELERATOR.

rate throttle valves may be used, arranged in the intake passage one above the other, one connected with the governor, the other connected to hand and foot controls, or simply to a hand control. The throttle controlled by the governor will remain fully open until the engine attains the speed for which the governor is set, when it will begin to close. A simpler and more common method consists in using only a single throttle valve to which the governor is connected by a slotted link. The engine is ordinarily under the control of the governor, but the driver may close the throttle independent of the governor.

Clutch and Brake Pedals—The conventional control for cars with sliding gear transmission comprises two pedals located on opposite sides of the steering post; the one on the left being the clutch pedal and the one on the right the

service brake pedal. The accelerator pedal or button is located slightly to the right of the brake pedal, for operation with the right foot. The clutch pedal usually is mounted on the shaft which carries the clutch-throw-out yoke and has its bearings in the clutch housing, while the brake pedal may be mounted on a shaft or stud supported by a bracket fastened to the frame side member, concentric (or nearly so) with the clutch pedal. Figure 11 shows a design (Marmon) in which both pedals are supported on a frame bracket, the clutch control embodying compound levers.

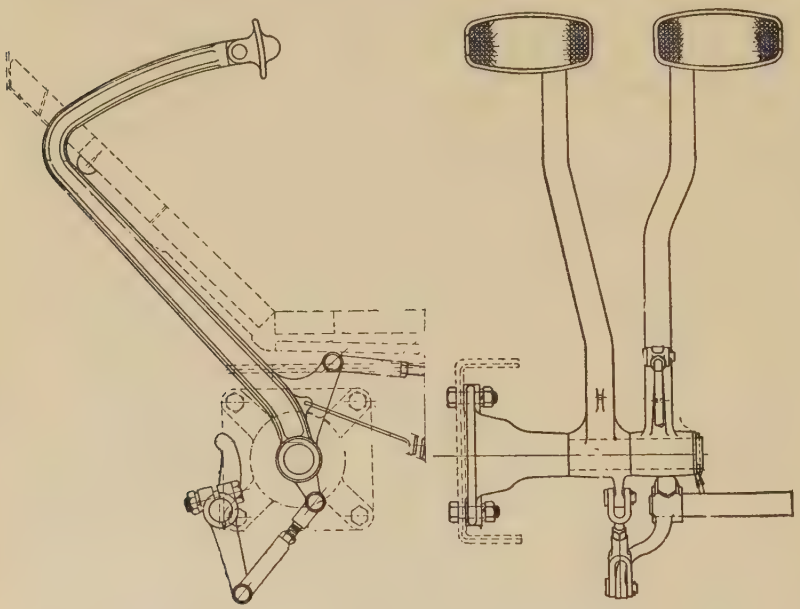


FIG. 11.—CLUTCH AND BRAKE PEDALS, SHOWING THEIR GENERAL FORM.

The ordinary one-piece pedal has a shank with a straight portion below the floor board and an arc-shaped section extending through an opening in the floor-board. If it is desired to have the pedal adjustable, the latter portion of the shank is often also made straight. Brake and clutch pedals vary in effective length from about 13 to about 16 in. The section near the hub generally is made with either an I-section or an oval section, with a section modulus around an axis parallel to the pedal axis, of from 0.08 to 0.12 in.[3] Near

the pad the section is made either oval or nearly rectangular, with an area of from 0.30 to 0.45 sq. in. If an I-section is used it generally extends from the larger end substantially to the junction between the straight and arc-shaped sections, while in Fig. 11 the I-section extends the whole length.

The pedal proper is generally drop-forged, but it is not convenient to forge the pad integral, and the latter is either bolted to the shank—which has the advantage that it permits of varying the angular relation between the two parts—or is welded on, generally by the flash-welding process. Preferably the whole of the pedal should be symmetrical with respect to a plane perpendicular to its axis and passing through the center of the hub. Sometimes it is found desir-

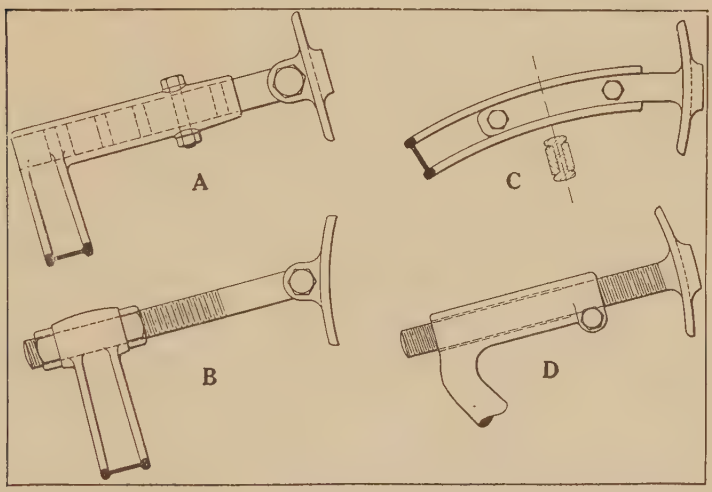


FIG. 12.—ADJUSTABLE PEDALS.

able to offset the pad considerably from the hub, in which case the torsional effect on the larger end of the shank must not be overlooked, and a solid section can be used to advantage.

Adjustable Pedals—The comfort of the driver is an important consideration in the design of a motor vehicle. It is evident that the distance from the front edge of the seat to the pedal pad which best suits a tall person is unsuitable for a person of short stature, and vice versa. Some makers are endeavoring to solve this problem by making the driver's seat adjustable in a fore-and-aft direction, while others pro-

vide adjustable pedals. Such adjustable pedals usually have shanks comprising two straight portions. Several forms of adjustable pedal are shown in Fig. 12. In design A the pad is secured to a round rod passing through a hole in that portion of the shank which extends through the opening in the floor. The rod is drilled with a number of transverse holes and may be secured in any one of a number of positions by means of a "through" bolt. In design B the shank of the pad is threaded and is clamped in a hub on the free end of the pedal shank by means of two nuts. Design D is similar, except that the shank of the pad screws through the hub on the free end of the pedal shank and is clamped in any desired adjustment by means of a clamp bolt. These various

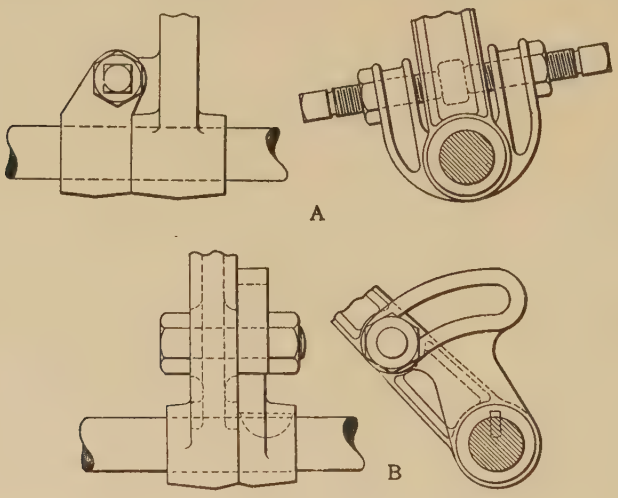


FIG. 13.—PEDALS ADJUSTABLE AT HUB.

designs with straight pad shanks have the disadvantage that they call for a somewhat longer slot in the toe-board, which is undesirable because it allows more cold air to enter the interior of the car in cold weather, and more smoke when the engine is hot. At C a design is shown which overcomes this objection. In this case the pedal shank is made of I-section, while the pad is made with two shanks fitting into the hollows of the I-section. The two parts are clamped together by two "through" bolts which may be passed through different holes in the pedal shank.

Adjustment for the pedals may be provided also at the

hub, and two methods of accomplishing this are illustrated in Fig. 13. In each case there are two hubs, one fixed to the shaft and the other free upon it. In design A the tight hub has two lugs with co-axial set screws through them, between the points of which is located a lug projecting laterally from the pedal. In design B the right hub is provided with a slotted sector which is cut with shallow radial grooves on one side, to which the pedal can be clamped in different positions.

Pedal Pads—Pedal pads are given many different forms —round and oval, square and rectangular—and in the cases of oval and rectangular shapes the long axis may be either in the plane of pedal motion or transverse thereto. In order to ensure uniform contact between the pad and the sole of the shoe for all pedal positions, the pad is usually made slightly convex toward the rear of the car.

To prevent the driver's foot from slipping off the pedal, the pad is either provided with diamond-shaped projections, formed in a press, or has a rubber pad clamped or vulcanized to it. Another method consists in providing the pad with an ear on one side, against which the driver may press his foot.

Some clutches are disengaged by pulling the throw-out collar to the rear, while others require it to be pushed toward the front. In the first case, if single-leverage control is used, the clutch pedal shaft is mounted above the clutch shaft and the throw-out yoke extends down from it, while in the second case the pedal shaft is mounted below the clutch shaft and the yoke extends upward from it. The brake pedal is usually free on the stud on which it is mounted, and a link of the brake hook-up connects to it, either at a lug above its fulcrum or at the end of an arm extending down from it.

It is important to provide rigid fulcra for both the clutch pedal and the brake pedal. Since the clutch pedal shaft usually has bearings in opposite sides of the very rigid clutch housing, this condition can be met easily, but in the case of the brake pedal this point deserves special attention. On heavy cars, application of the service brakes for an emergency stop usually requires the full strength of the driver. The pressure of the pedal on the fulcrum will be much greater, and if the fulcrum yields materially under this pressure, the useful pedal range is correspondingly reduced.

Location of Control Levers—In the conventional modern motor vehicle (American), in which the steering post is located on the left-hand side, there are two control levers at the center of the driver's compartment, the gear-shift lever

and the emergency brake lever. The advantage of having both of these levers located at the center is that the driver can then operate them with his right hand, which is usually the most dexterous. Moreover, both levers are then most advantageously located from the standpoint of connections. The gear-shaft lever generally is mounted in the vertical center plane of the car, on the cover of the gearcase, and requires no linkage whatever, while the brake lever either controls a transmission brake, which is located at the center of the car, or connects to a cross-shaft from which the force of brake application is transmitted to wheel brakes through side links, and if connection is made to the cross-shaft near its middle, its two sections will be of nearly equal torsional rigidity and therefore will yield about equally when equal moments are applied to them.

There is a disadvantage in center control, however, in that the levers interfere somewhat with ingress to and egress from the driver's seat, the degree of interference depending upon the proportions of the driver's compartment. Since in cars with four-wheel brakes the so-called emergency brake is used almost only for parking, some designers prefer to locate it to the driver's left.

With the steering post located at the right side, as customary in England and continental countries, the control levers may be located centrally, in which case the driver must operate them with his left hand, or they may be located on the right-hand side. In the latter case it is advisable to provide the gearcase cover with a lateral extension containing an outboard bearing for the shift-lever shaft. Formerly this bearing was often mounted on the frame side member, but this gave a good deal of trouble from the shaft jamming due to frame distortion.

Gear-shift Lever—Change gears now are controlled by cane-type levers which have a universal mounting either in a pedestal formed on the cover of the gearcase or in a cover on the clutch housing, this mounting forming the fulcrum of the lever. In order that the lever may interfere as little as possible with access to the driver's seat, the point where it projects through the floor boards should be as far forward of the seat as possible, and this has led to mounting of the lever on the clutch housing cover, which necessitates that the slider bars of the transmission extend into the clutch housing. A variety of mountings for the gear-shift lever are shown in Fig. 14. The lever itself generally is made of solid round section, tapering from the fulcrum to the ball handle. Near the fulcrum the diameter is made equal to the face width of

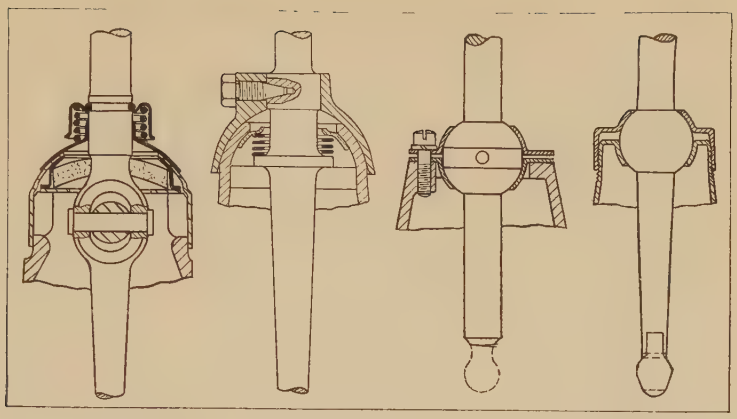


FIG. 14.—METHODS OF MOUNTING CANE-TYPE OF GEAR-SHIFT LEVER.

the constant-mesh gears and near the ball handle equal to 0.6 this value. If a ball support is used its diameter is generally about 2.5 times the face width of the constant-mesh gears.

These levers are fitted with ball handles of some non-metallic material such as bakelite, of from 1½ to 2 in. diameter. The handles are molded with metallic inserts to screw over the threaded ends of the levers, and the inserts have been standardized by the S. A. E. For passenger cars the thread on the upper end of the lever is made ⅜ in.–24, and for motor trucks ½ in.–20, the length of the thread being ⅝ in. in both cases. Figure 15 shows two typical ball handles. It also shows a finger latch on the shift lever on the right, used to unlatch the reverse position. The proper location of the ball handle with relation to the front edge of the seat when the gear is in neutral is shown in Fig. 20 and this and the location of the fulcrum determine the length of the lever.

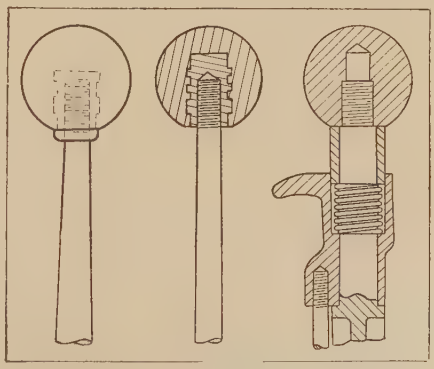


FIG. 15.—BALL-HANDLES FOR SHIFT-LEVER.

Gearshift Positions—Gearshift positions for both three-speed and four-speed transmissions have been standardized by the S. A. E., and these standards are now in use on practically all American passenger cars and on most trucks, while in addition they are being adopted abroad. The standard positions for the two types of transmission are shown in Figs. 16 and 17. In the four-speed standard the position of the reverse is left open. It is usually to the right of the low and second positions, and entrance to it is barred by a "reverse lock-out," such as was illustrated in the chapter

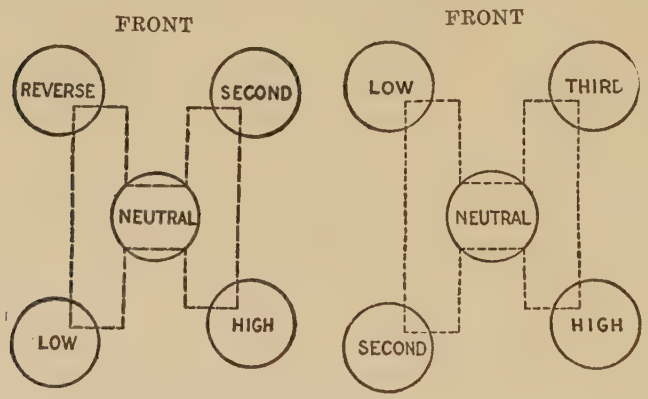


FIG. 16 (LEFT).—STANDARD SHIFT-LEVER POSITIONS FOR THREE-SPEED GEAR.

FIG. 17 (RIGHT).—STANDARD SHIFT-LEVER POSITIONS FOR FOUR-SPEED GEAR.

on Sliding Gear Transmissions. One thing worth noting in connection with the four-speed-gearshift-positions standard is that if the vehicle is habitually started in second instead of in low gear, the operation is exactly the same as that of a three-speed gear.

Emergency Brake Lever—The emergency brake or parking brake is operated by means of a hand lever which is provided with a latch so the brake can be set in the "on" position. These levers are generally drop forged and given an oval section for passenger cars and an I-section for trucks and buses. The lever is fulcrumed on a bracket cast on the transmission case or its cover, and its handle when the brake is "off" must occupy the position shown in Fig. 20. In the Ford Model A car, in order to obtain a lever of great rigidity and still of light weight, it is made of steel pressings

which are welded together to give a hollow section. The levers vary considerably in length. In the case of brake levers for heavy trucks and buses the section modulus near the fulcrum can be made 0.12–0.14 in.^{3} and near the handle 0.05–0.06 in.,^{3} while for passenger cars it can be made 0.06–

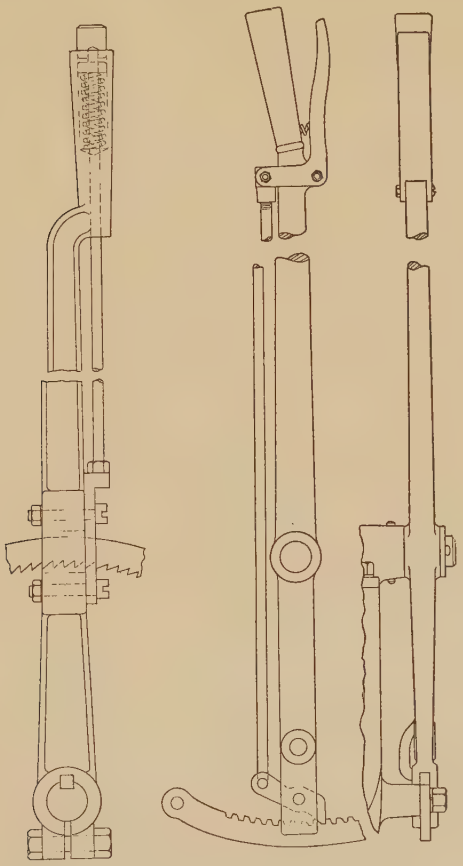


Fig. 18 (left).—Thumb-Latch Lever.
Fig. 19 (right).—Spoon-Latch Lever.

0.007 in.^{3} near the fulcrum and 0.02–0.025 in.^{3} near the handle.

There are two types of latches for levers, the thumb latch and the spoon latch, and levers with both types are illus-

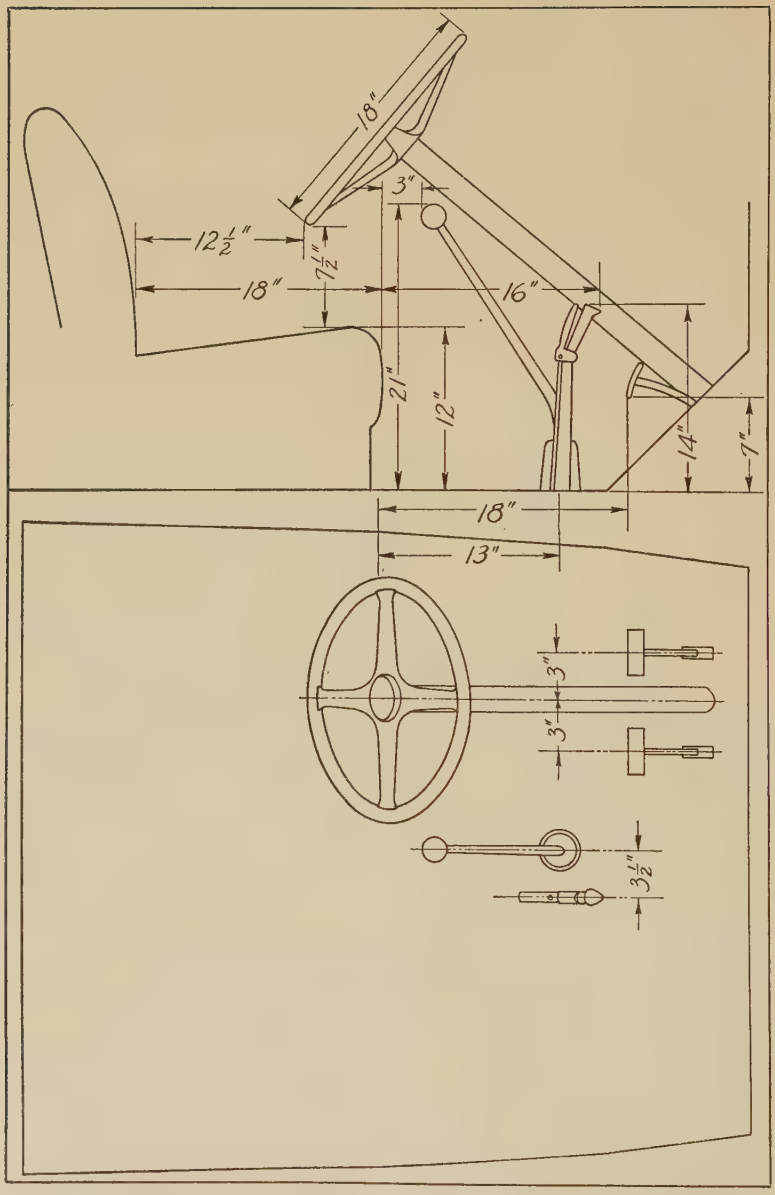


Fig. 20.—Proper Locations for Passenger-Car Control Devices.

trated in Figs. 18 and 19. The operation of these latches is self-evident and need not be described. Emergency brakes are now always applied by pulling the lever back, and the spoon latch must therefore be on the rear side of the handle. The handle is made from 3 to 4 in. long and of round section tapering from $\frac{5}{8}$ in. diameter at the lower to $\frac{7}{8}$ in. at the upper end, or else it may be made substantially cylindrical but with a ledge on the forward side to prevent the hand from slipping off. Frequently the handle is bent forward so it will be substantially perpendicular to the extended arm when in mid-position.

CHAPTER XV

Frames and Their Brackets

Frames for motor vehicles are almost exclusively made of sheet-steel pressings. Wood sills have been used by one American manufacturer of passenger cars, and have a slight advantage over pressed-steel frame members made of carbon steel in respect to strength in proportion to weight, but the quality of wood necessary for such frames is difficult to obtain, and, moreover, wood construction s suitable only for straight sills, while modern passenger car design calls for a "kick-up" over the rear axle and in some instances also for a drop back of the front spring. Frames of rolled channel sections formerly were much used for trucks. The material of these sections has better physical properties than that used for pressed frame sills, but since a rolled sill of necessity has the same section throughout, while the bending moments which it must withstand vary from practically nothing to a maximum, there is a considerable waste of material in such frames. Pressed-steel frames therefore have practically replaced rolled-steel frames for trucks.

The material mostly used for frames is cold-rolled, open-hearth steel of the straight carbon type, with either 0.20 per cent or 0.25 per cent of carbon content. Various alloy steels have also been used for frames, notably a chrome-nickel steel No. 3230, and a molybdenum steel. These steels are heat treated, for without heat treatment the gain in elastic limit is not sufficient to warrant the additional cost of the material.

There was really more incentive for using alloy steels for frames during the earlier period of automobile history, when most of the cars built were of the open type, and frames were habitually made exceedingly light. At that time frame breakages were not at all uncommon. When the four-door-type of body was introduced, considerable trouble was experienced with the operation of the doors, owing to deflection of the frame, which was usually a maximum at about the rear edge of the front door. This and the later great increase in

popularity of closed bodies, which type of body is subjected to great stresses if the frame deflects excessively under load, made it necessary to design frames more with a view to rigidity than to breaking strength, and the cheaper carbon steel gives the same rigidity as the more expensive alloy steel. As a rule, if the frame (of a passenger car) is sufficiently rigid to protect a closed body against inordinate stresses, it meets all requirements of strength.

The most widely used frame material is open-hearth carbon steel of about 0.20 per cent carbon content. In the annealed condition such steel has an elastic limit of about 40,000 lbs. per sq. in. Steel with a somewhat higher carbon content, about 0.25 per cent, is also used, and has a somewhat higher elastic limit, but is not as malleable as the low-carbon product. Cold rolled steel is used mostly, the chief reason being that it has a natural bright finish as it comes from the mill. The physical properties of three frame materials are given in Table I below. Sheet for frames is measured by the U. S. sheet metal gage, and is furnished in thicknesses of 0.125, 0.140, 0.156, 0.187 and 0.250 in.

Frame Sections—The side rails of automobile pressed-steel frames are invariably made of channel section, with the open side turned inward. Most of the cross-members used are also of channel section, although almost every frame has either one or more tubular cross-members to add to its resistance to twisting moments. The side rails are the most important parts, since they form beams of considerable span and are therefore subjected to great bending stresses. They also constitute the bulk of the weight of the frame. The height of the section is made constant over a certain length at the middle of the side rail, usually about one-third, and decreases uniformly toward both ends.

Table I.—Physical Properties of Frame Steels

Material	Elastic Limit, Lbs. per Sq. In.	Reduction of Area, Per Cent	Elongation in 2 In., Per Cent
S.A.E. steel No. 1020 (natural)	40,000	60	30
S.A.E. steel No. 1025 (natural)	45,000	55	30
S.A.E. steel No. 1025 heat treated	60,000	64	24
S.A.E. steel No. 3230 heat treated	85,000	65	22

Types of Frame—Motor car frames must be made narrow in front, in order to give a sufficient steering lock. In

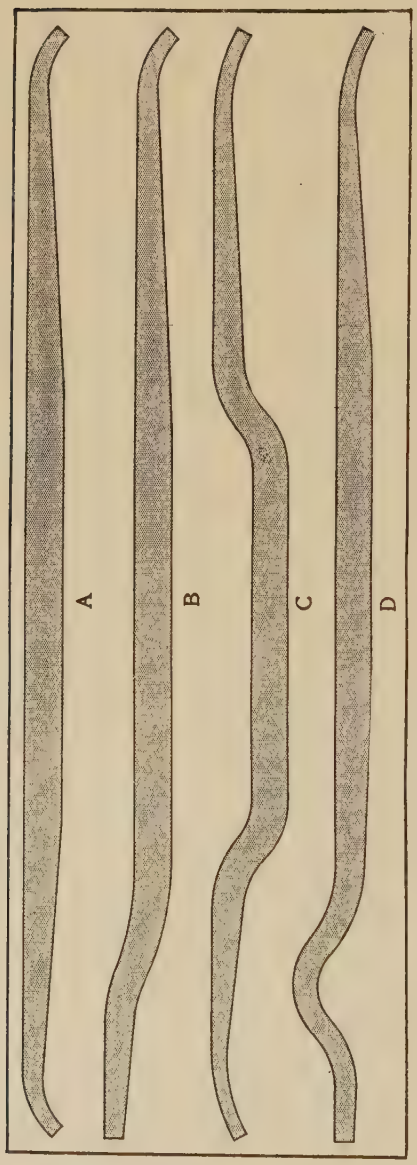


FIG. 1.—VARIOUS FORMS OF FRAME RAIL IN SIDE VIEW.

the rear they must be made wider, as the body at the rear will be much wider than the permissible frame width at the front, and the body sills must be bolted either directly to the frame side rails or to brackets riveted to them on the outside. From the manufacturing standpoint the straight frame (one with parallel side members) has advantages, and a good many truck and bus frames are made that way.

In the earlier passenger cars the frames generally had an offset (or insweep) near the rear end of the front springs, the narrow forward portion and the wide rear portion both having the side rails parallel. The objection to this design is that it reduces the resistance of the frame to vertical bending moments materially. There are various methods of minimizing this objection, as, for instance, by widening the flange for some distance past the point where the offset begins, so the inner edge of the flange continues in a straight line from the narrow section; by inserting a straight reenforcing channel in the side rail at the narrow part and extending it to a cross-member at the wider part, and, finally, by using a frame truss with the strut at the middle of the offset. However, all of these schemes involve additional cost and weight, and offset frames are now only rarely used.

The preferred type of frame for passenger cars has straight side rails (viewed from above) which form a trapeze, the rails being closer together at the front than at the rear. This type has the advantage of maximum simplicity of frame rails, combined with maximum strength for a given weight of material. A disadvantage is that, owing to the inclination of the side rails to the car axis, certain brackets and other parts riveted to the frame may have to be machined at an angle.

In a second type the frame has parallel sections at the front and rear and tapers most of the distance between springs, while in a third type the greater part of the length from the forward end tapers, while the rear portion is parallel. Four types of side rail are shown in side view in Fig. 1, A being the ordinary straight side rail; B, a rail with a single drop, at the rear end; C, a rail with double drop, as now frequently used for passenger cars of low floor height; and D, a rail with an arch or a kick-up over the rear axle.

Width of Frame—Widths of frames for cars with the standard 56-in. tread are limited to 30–32 in. at the front axle, as otherwise the steering lock will be insufficient. At the extreme rear the width of frames of passenger cars of standard width or slightly over is generally between 45 and

48 in. Frame widths for motor trucks have been standardized by the S. A. E. as follows:

Straight Frames—

Speed wagons, 32 in.; ¾–2½-ton, 34 in.; 3 ton and over, 38 in.

Offset Frames—

¾–2½-tons, front, 31 in.; rear, 34 in.

Calculation of Side Rail Section—Each frame side rail constitutes a beam which is supported at four points, the points of spring attachment, as shown in Fig. 2. The reactions R at opposite ends of each spring will be equal, and the resultant of these two reactions acts on the frame midway between spring eyes or directly above the axle. In a passenger car of standard design the weight on the frame is distributed more or less uniformly from a point substantially above the front axle to a point a little behind the rear axle, and, therefore, we will not be far wrong in considering the frame side bar a beam supported at two points, directly above

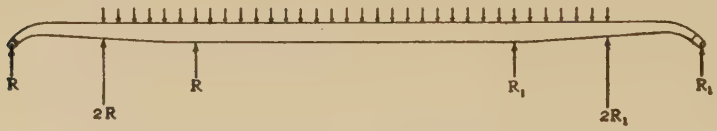


Fig. 2.—Diagram of Load and Reactions on Frame Rail.

the axles, and carrying a uniformly distributed load between points of support.

Let l = the distance between supports (wheelbase), and W the weight carried by each frame rail, then the maximum bending moment, which occurs midway between supports, is $Wl/8$. It is evident that the modulus of the section at the point of maximum bending moment should be proportional to this moment. Calling the necessary factor of safety f, then

$$\frac{Wl}{8} = \frac{ZL}{f},$$

where Z is the section modulus and L the elastic limit of the material. Just what value should be given to f cannot well be determined from first principles. The above equation may be transposed to read

$$Z = \frac{Wlf}{8L}.$$

The weight W on one frame rail is proportional to the total weight of the car with load (W_1). Hence we may write

$$Z = \frac{aW_1lf}{8L} = \left(\frac{af}{8}\right)\frac{W_1l}{L}.$$

Denoting the expression $\left(\frac{af}{8}\right)$ by c we have

$$Z = c\frac{W_1l}{L}.$$

Frames of a considerable number of passenger cars were analyzed, and the value of c was found to vary from about 0.14 in the lighter, cheaper models, to about 0.24 in the heavier, higher-grade cars, the average being 0.20. The analysis in each case was based on the "fully-loaded" weight of the heaviest model of the line. In all passenger cars the distribution of weight is very nearly the same. The radiator, the most forwardly located part of the load, usually sets directly over the front axle, while the rear part of the body slightly overhangs the rear axle. Frames for cars with such a weight distribution will be of adequate strength if the section modulus of the side rails is made

$$Z = 0.20\frac{W_1l}{L}.$$

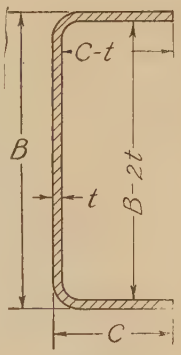


FIG. 3.

In motor trucks and buses the weight distribution with respect to the two axles is quite different, and the constant in the equation for the necessary section modulus of the frame side rails therefore must be given a different value. Usually in a fully loaded truck about three-fourths of the total weight is on the rear axle, the pay load itself being practically centered on this axle. For such vehicles the constant in the equation for the section modulus can be made 0.12.

The section modulus of a channel section (Fig. 3) is equal to

$$Z = \frac{CB^3 - [(C - t)\,(B - 2t)^3]}{6B}.$$

In Table II are given section moduli for sections within the size range employed for motor vehicle frames.

Table II.—Frame Channel Section Data

B	*C*	*t*	*Z*	Add.*
4	1.5	0.125	0.98	0.117
4	1.5	0.140	1.08	0.130
4	1.5	0.156	1.19	0.143
4	1.5	0.187	1.40	0.170
4.5	1.5	0.125	1.15	0.133
4.5	1.5	0.140	1.27	0.148
4.5	1.5	0.156	1.41	0.163
4.5	1.5	0.187	1.76	0.193
5	1.5	0.125	1.33	0.149
5	1.5	0.140	1.49	0.165
5	1.5	0.156	1.63	0.183
5	1.5	0.187	1.93	0.217
5	1.5	0.250	2.45	0.281
5.5	1.75	0.125	1.69	0.164
5.5	1.75	0.140	1.88	0.182
5.5	1.75	0.156	2.06	0.202
5.5	1.75	0.187	2.45	0.240
5.5	1.75	0.250	3.14	0.313
6	1.75	0.125	1.91	0.180
6	1.75	0.140	2.36	0.200
6	1.75	0.156	2.54	0.222
6	1.75	0.187	2.78	0.254
6	1.75	0.250	3.56	0.344
6.5	1.75	0.125	2.16	0.195
6.5	1.75	0.140	2.38	0.217
6.5	1.75	0.156	2.64	0.241
6.5	1.75	0.187	3.17	0.287
6.5	1.75	0.250	4.01	0.375
7	2	0.125	2.61	0.211
7	2	0.140	2.83	0.235
7	2	0.156	3.19	0.261
7	2	0.187	3.78	0.310
7	2	0.250	4.88	0.406
8	2	0.125	3.14	0.242
8	2	0.140	3.50	0.270
8	2	0.156	3.88	0.300
8	2	0.187	4.21	0.358
8	2	0.250	5.96	0.469

* Values to be added to *Z* for each $\frac{1}{4}$ in. above that given in the column for *C*.

Bending Moments and Shearing Forces on Side Rails—It is, of course, possible to calculate the bending moments at various points along the side rails if the weights of the different components of the chassis and of the body and load,

and their exact location are known, and a diagram of bending moments and shearing forces can then be drawn. Such a diagram for a 4-ton truck, with loading platform centered over the rear axle, is shown in Fig. 4. In this case there is a point of zero bending moment some distance ahead of the front axle, and the bending moments are *nil* also at both ends of the frame. In a passenger car, on the other hand, in which practically the whole of the load supported by the frame is located between the two axles, the bending moment is zero at both ends of the frame and a maximum at a point near the middle of its length, the curve of bending moments being of the general form of a parabola. Such diagrams of bending moments and shearing forces are useful in indicat-

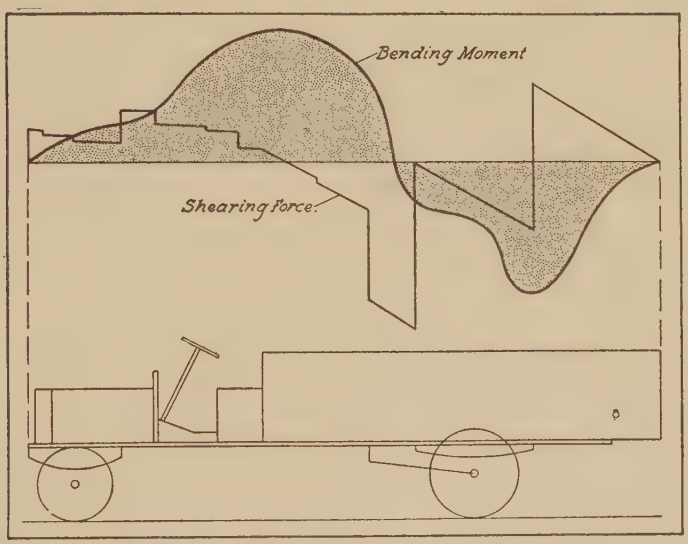


Fig. 4.—Diagram of Bending Moments and Shearing Forces on Motor Truck Frame Rail.

ing points of maximum stress; but it is impossible to obtain a high degree of accuracy in them. If the body rests directly on the frame side rails it is necessary to assume that the weight of body and useful load is distributed over the frame uniformly over the length of support, but the actual distribution may be quite different.

Proportions of Frame Parts—At one time many parts of pressed steel frames were standardized by the S. A. E., but with the one exception of truck frame widths these standards

and recommended practices have all been cancelled. Constant progress in the design of other units calls for changes in frame design, and the standards appear not to have been of any particular help. They were mainly of the nature of preferred numbers. Thus the radius of the lower edge of the frame rail at the front spring horn varied in multiples of 4 in., the drop at the forward end of the rear spring varied in even inches, the radii of the double curve at the rear drop varied in multiples of 10 in., etc.

Channel pressings are made with inside corner radii of 3/16 in. for depths of section below 5 in. and 1/4 in. for sections deeper than 5 in. Rivets of 5/16 in. and 3/8 in. diameter are used for hot-riveting of frames, and punched holes for these rivets are made 11/32 and 13/32 in. in diameter, respectively. The usual spacing of rivets for frame members is about 1½ in.

Stresses on Frames—In the past there have been two schools of frame design, one aiming to make the frame as rigid as possible within the limits imposed by considerations of weight, so that its deformation under road shocks would be limited and all parts carried by it would be protected against undue stresses due to such shocks. The other school aimed at making the frame flexible, so it would yield under road shocks and would not be unduly stressed by them itself. At present almost every designer of passenger cars aims at maximum rigidity, as without this it is difficult to prevent closed bodies from developing squeaks at an early stage of their life.

Frames may be distorted in two ways. We may imagine the forward end of one side rail to strike an obstruction, such as a fence post, wall, etc., which would displace it in the direction of its length relative to the other side rail. This is called "weaving." On the other hand, we may imagine one wheel to pass slowly over an obstruction of considerable height, so that all or practically all of the load at that end of the vehicle is thrown onto it. This results in what is known as twisting of the frame. It is the object of the cross-members to prevent such "weaving" and twisting, in addition to which they serve the purpose of supporting certain of the components mounted on the frame.

Frame Cross-Members—Twisting is best reduced by the use of tubular cross-members, and this type is much used at the front and rear ends, and sometimes also at the middle. For maximum efficiency the tube must be of large diameter and thin-walled, and it is usually riveted or welded to a flange of square or oval shape, which in turn is riveted to

the web of the side channel. Tubes of 1½ in. outside diameter are often used for frames of the lighter passenger cars, and 2 in. for the heavier ones, but tubes as large as 4 in. are sometimes used. As a rule, the tubular form is preferred for those cross-members to which no chassis components have to be fastened; naturally it is more difficult to mount such parts on a tube than on a pressing, which latter can be given any desired shape.

Location of Cross-Members—At the forward end a cross-member is required to carry the radiator. As a rule this is a channel-shaped pressing with the open side down, and dropped in the center. The reason for the drop is that the

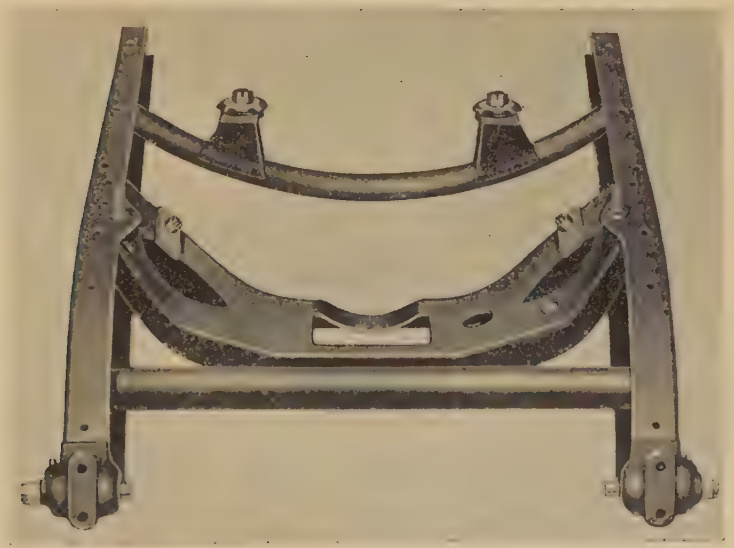


FIG. 5.—FORWARD END OF PASSENGER-CAR FRAME (Nash).

radiator must extend down lower than the top of the side rails. Frequently this cross-member also supports the forward end of the engine.

This is illustrated in Fig. 5 which shows the forward end of the Nash passenger-car frame. The forward cross-member is made of considerable width and really divides into two parts in the direction of the car axis, the forward section being shaped to conform to the radiator outline, while the rear section has pads on which the crankcase feet rest. Ahead of this pressed steel cross-member there is a tubular cross-

member of large diameter, designed to give additional rigidity to the forward end. Flexibility of the front end has been found to promote front end vibration or wheel wabble, and various means have been resorted to to combat it. In the Auburn frame for instance, the front end of which is illustrated in Fig. 6, the side channels of the frame are reenforced from the dash forward by inserting another channel in each of them. This frame also has a front cross-member of peculiar shape, which may be regarded as two angle sections side by side. It is referred to as a pan bar, the under-pan forming a joint with its rear section.

Engine Arm Brackets—Next come the engine supporting members, which in most cases also act as frame cross-members. The engine supporting arms generally rest on brackets riveted to the frame side rails, so that the engine can be lowered into place from above. It has become the custom to support the engine on rubber blocks to prevent the transmission of vibrations from it to the frame and body, which, however, makes the engine arms or carriers rather ineffective

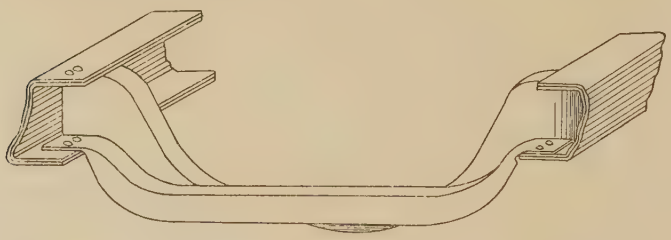


Fig. 6.—Front Cross Member and Pan-Bar (Auburn).

as cross-members. This can be obviated by using a rigid cross-member and mounting the engine on cushions on it, as in the Nash design, Fig. 5. Here there is a tubular engine bearer in the form of an inverted arch, on which the rear end of the engine is supported.

There is usually a cross-member a short distance back of the transmission housing and another close to the rear-spring-front brackets, or slightly further back, at the drop in the frame. If these cross-members are steel pressings they are generally given a deep section, especially at the middle, which adds greatly to the resistance of the frame to twisting stresses. Fig. 7 illustrates a double cross-member for the middle of the frame which has been used in several variations by the German Daimler company. It is substantially X-shaped, each

member having a channel section. In an earlier passenger car with separate transmission the open sides of the channels came together, and the channels were riveted to inserts so as to form a box-section on which the ball-housing for the forward end of the propeller shaft tube was mounted. In the design here shown the closed sides of the channels lie together and a mounting for an intermediate bearing for the propeller shaft is bolted to them. It is readily seen that a cross-member of this type must give great resistance to weaving, and if provided with a box-section at the middle, also against twisting.

In some cases the cross-member behind the transmission is pressed with a large round or oval hole through it, through which the propeller shaft passes, and it then acts also as a safety device, preventing the shaft from falling to the ground

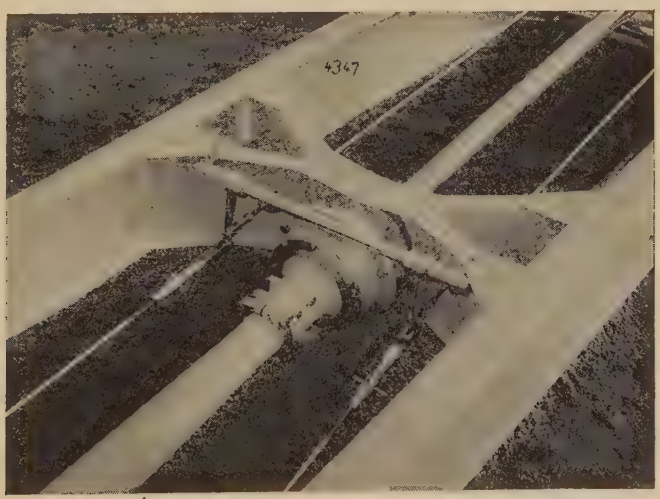


Fig. 7.—X-Type Central Cross Member (Mercedes).

in case the forward universal joint should come loose, which is not an impossibility. This type of cross-member is shown in Fig. 8, which illustrates the center section of a Buick frame. This same view shows a number of angle brackets riveted to the outside of the frame rails for supporting the body. As a rule, the body sills rest directly on the frame side rails, but the use of such body-mounting brackets makes it possible to reduce the width of the frame at the rear, which is often desirable.

If a pressed steel cross-member is used at the rear drop in the frame, it can be so widened at the ends that these may be riveted to the top flange at the high part of the frame and the bottom flange of the low part, as illustrated in Fig. 9, which shows the rear end of a Buick frame. This spreads the connection between cross-member and side rail over a considerable distance in both the horizontal and vertical planes, and therefore tends to give general rigidity.

In passenger cars the rear ends of the two side rails are usually connected by a wide plate which serves as a shield or guard for the fuel tank and sometimes has the spare tire carrier fastened to it at its center. As illustrated in Fig. 10, this follows the upper flange of the side rails over most of its width, and then jumps down to the lower flange, which it may also follow for some distance. The two edges of this

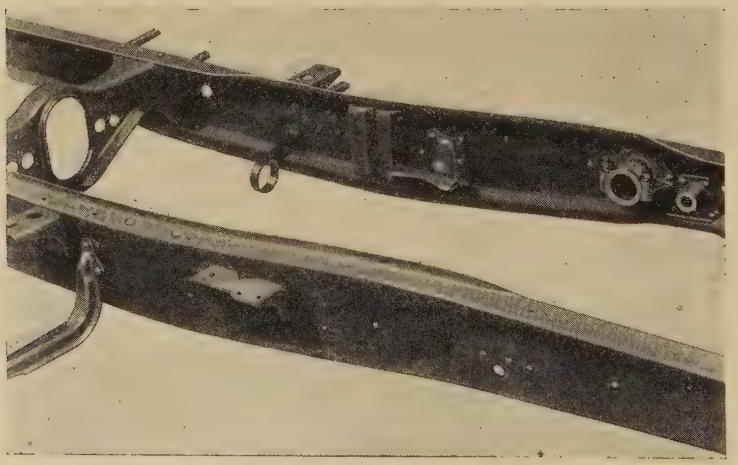


FIG. 8.—CENTRAL PART OF PASSENGER-CAR FRAME (Buick).

plate also are a considerable distance apart in both the horizontal and vertical direction, and therefore tend to give great stiffness. Nevertheless, many passenger car frames in addition have a tubular cross-member directly back of it, to add to the torsional stiffness.

In designing drop bars, the radii of the outlines should be made as large as possible, as short curves are hard to draw. The flanges should be made of equal width, or nearly so, and the ends of the bar should preferably be so designed that the flanges are trimmed to the same length. Stock will

be economized if an integral gusset is provided for on the lower flange.

Joints between Frame Members—The parts of pressed steel frames are joined by riveting (bolting in rare instances), either separate or integral gussets being used (A and B, Fig. 11). Two methods of riveting are in use, viz., cold riveting and hot riveting. In testing a hot-riveted joint under tension in the plane of contact, as the tension assumes a certain definite value, there is a sudden increase in extension. This is due to the fact that up to this point the tension is resisted by the contact friction, which in a hot riveted joint is very considerable, because the rivet in cooling draws the two parts together with great force; and to the further fact that the

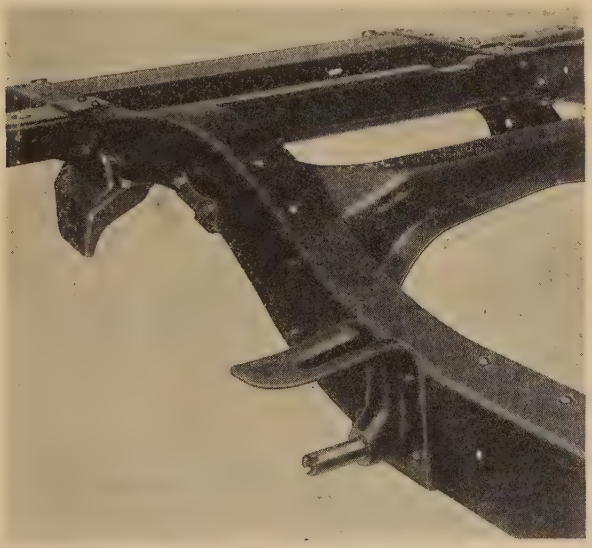


FIG. 9.—CROSS MEMBER AT REAR DROP (Buick).

rivet, also because of its contraction in cooling, does not entirely fill up the hole. With cold riveting the hole is completely filled by the rivet, but, on the other hand, the surfaces are not applied to each other with as great force. It seems that hot riveting, on the whole, has proven the most satisfactory and is now in general use. Two sizes of rivets are used, 5/16 and 3/8 inch. The holes for hot riveting for these two sizes are made 11/32 and 13/32 inch, respectively, and are spaced about 1½ inches. In riveting brackets to frame

members, three rivets are often used. All rivet holes weaken the frame, but the weakening effect varies greatly with the location of the holes. A hole at the middle of the web has but little effect, but the opposite is true of a hole in one of the flanges. In this connection it is well to remember that the lower flange is under tension and the upper under compression, the unit stress being the same in both, and since the compressive strength of frame materials is somewhat greater than their tensile strength, it is advisable to put rivet holes in the upper rather than in the lower flange, where this can be done just as well.

Spring Brackets—The chassis frame is carried on the springs through the intermediary of spring brackets. At the front semi-elliptic springs are used, as a rule, which require

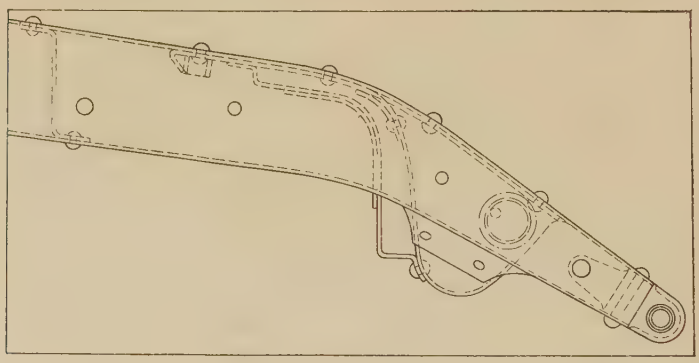


FIG. 10.—FUEL-TANK GUARD PLATE AT REAR END.

a bracket at either end. The forward bracket is generally made in the form shown in Fig. 12. It fits into the downwardly curved forward end of the frame channel to which it is secured by one rivet in the vertical plane at the extreme forward end of the channel, and three or more horizontal rivets. In the cheaper cars and in commercial vehicles this bracket is usually made in the form of a plain forked connector, but in the design illustrated the spring eye is surrounded by a shroud which extends a little below the axis of the spring bolt and completely encloses the spring eye. This makes for a neater appearance than an open-forked bracket. The eye bolt can be held from rotating in the bracket either by a small pin extending into its head from underneath and into the prong of the fork, or by a small key.

The rear ends of semi-elliptic front springs are connected

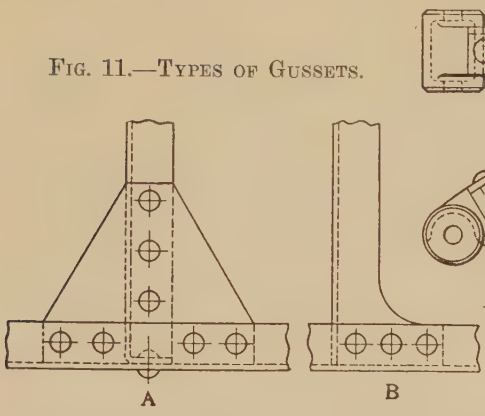


Fig. 11.—Types of Gussets.

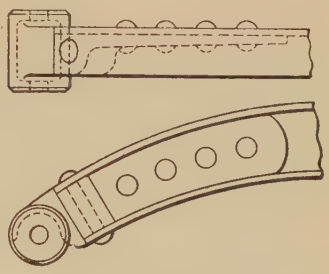


Fig. 12.—Front-Spring Front Bracket.

to the frame brackets by shackles, and these shackles may work either under compression or under tension, the brackets being designed accordingly. Simplicity of construction is in favor of shackles under compression, and these are now generally used, even on the most expensive cars. Figure 13 shows two forms of front spring rear brackets for shackles working under compression. That shown at A does not require any rivets through the flange of the frame rail, and therefore may be considered the better construction, although at this point of the frame rail the strain on the material usually is not very great.

The rear springs at their front end are either pivoted or shackled to their brackets. Two designs of brackets for this part of the car are illustrated in Fig. 14. The one shown

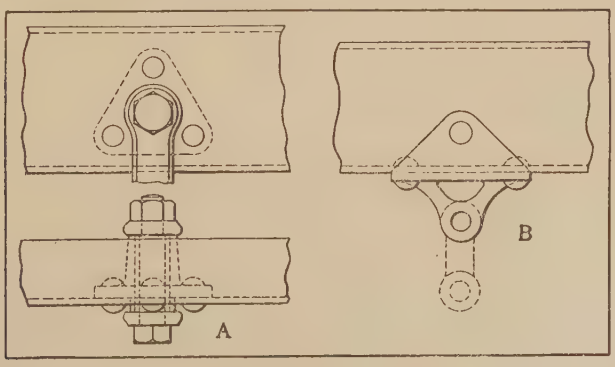


Fig. 13.—Front-Spring Rear Brackets.

at A is a plain fork and is secured to the frame by three rivets, one of which passes through the lower flange. The design shown at B is of the shrouded type, which gives a somewhat neater appearance if the front end of the rear spring is exposed to view. Generally, however, it is covered with a shield, and an open type of bracket is used.

If semi-elliptic springs are used at the rear the frame rail may be curved downwardly, the same as in front, and provided with a bracket. A bar is then run through the eyes of the bracket on opposite sides of the frame, whose ends, somewhat reduced in diameter, serve as the spring shackle bolts. This practice is more or less prevalent in Europe.

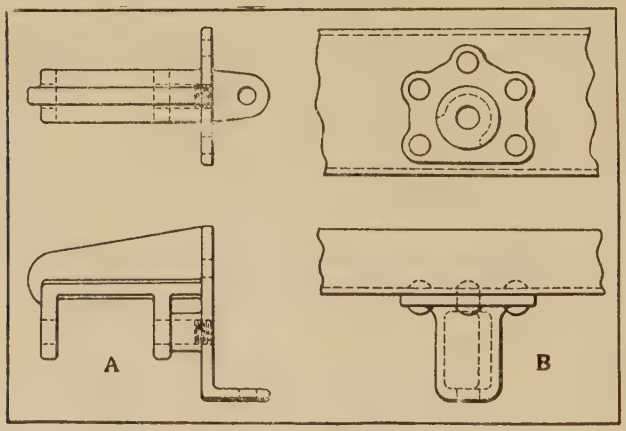


Fig. 14.—Rear-Spring Front Brackets.

Spring brackets for motor trucks differ from those for passenger cars on account of the difference in frame construction and because neat appearance is not such an important factor. Figure 15 illustrates two designs of truck spring brackets. That shown at A is a front bracket and is riveted to the front corner of the frame. However, the tendency in American design is to have the frame overhang the springs in front, so the bracket comes underneath the frame. A bracket of the type shown at B is used at the rear end of the front springs and also at the shackled end or ends of the rear springs.

Bumper Pads—Front spring front brackets now are often made with integral bumper pads, as illustrated in Fig. 5. In this connection it is worth remembering that the standard height from the ground of the center of front bumper bars is 18 in., and of rear bumper bars, 19 in. (both plus or minus ⅜ in.), this height being measured with the car ready for the road but without passenger load.

Running-board Brackets—Running-board brackets standardized by the S. A. E. are widely used. These are made of pressed steel and have the form and dimensions shown in Fig. 16. Only three heights are specified, but by varying the position of the base of the bracket on the web of the side rail, the vertical distance from the bottom of the side rail to the top of the bearing part of the bracket can be

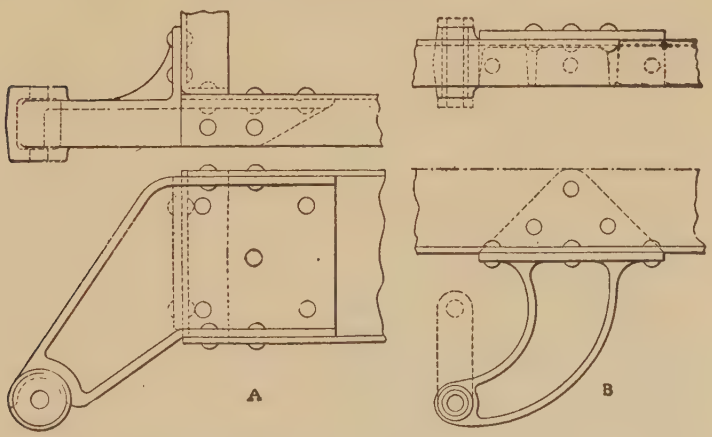


FIG. 15.—MOTOR-TRUCK SPRING BRACKETS.

made anything desired within a wide range. For passenger car running boards, usually three of these brackets are used on each side, while two will suffice for the steps of motor trucks.

Frame Outriggers—Reference has been made already to the fact that in some passenger cars the body, instead of setting directly on the frame, is supported on brackets riveted to the frame on the outside, the body then overhanging the frame side rails. There is a much larger overhang in the case of motor buses, and bus-frame side rails usually have

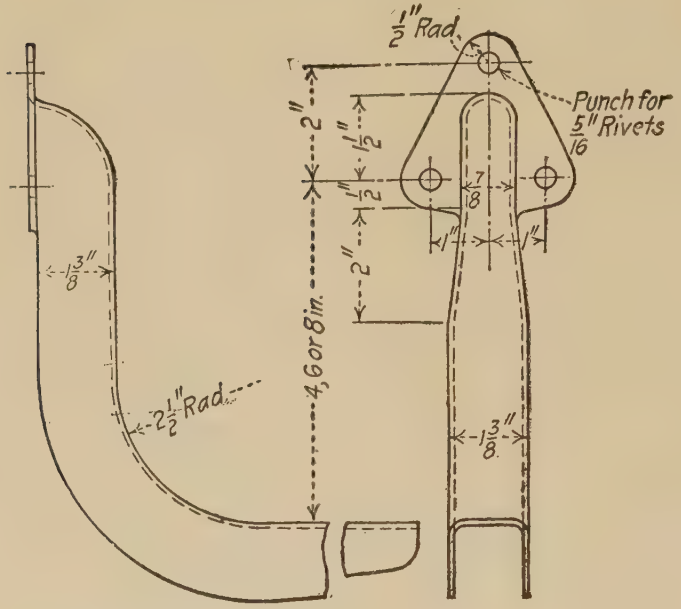


FIG. 16.—S. A. E. STANDARD RUNNING-BOARD BRACKET.

outriggers riveted to them on the outside to support the body. One design of a bus-frame outrigger is shown in Fig. 17.

Commercial Vehicle Bumpers—While the bumpers of passenger cars are manufactured by specialists, those of commercial vehicles, and especially of motor trucks, are generally made by the vehicle manufacturers themselves, or by the frame manufacturers. There is therefore less uniformity in these bumpers. The object of bumpers, of course, is to take the shock of collision and transfer it directly to the frame, thus protecting frail parts at the front of the vehicle, such as lamps, radiator, etc. One manufacturer bends the frame

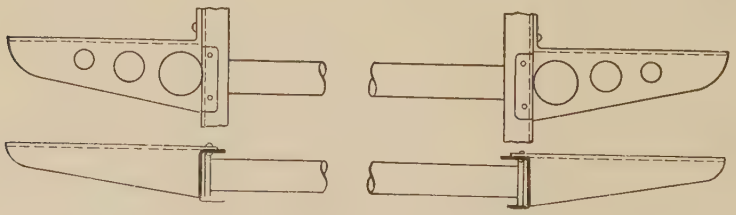


FIG. 17.—FRAME OUTRIGGERS FOR BUS BODIES.

FIG. 18.—MOTOR-TRUCK BUMPER.

side rails inward in a curve, so they meet at the middle of the chassis, where they are joined together by a plate riveted to them. Another uses a sort of arch formed of angle iron which is riveted to the frame side rails. In some cases a heavy tube is used, held in eyes of brackets projecting forward from the frame side rails. The neatest arrangement is undoubtedly a pressed-steel channel-section bumper, with the open side of the channel toward the rear, which is bolted to pads formed on the front spring horns or riveted to the ends of the side rails. Since the web of the channel does not add materially to its strength, it can have sections stamped out, which also improves the appearance.

Fender Brackets—Fender brackets are preferably bolted to the frame, as fenders are easily injured and may have to come off. It is also desirable to remove at least one of the fenders if much work has to be done on the engine without removing the latter from the chassis. Figure 19 shows the method of bolting the brackets for the front fenders to the frame in the Pierce-Arrow truck. These brackets are malleable castings. Front fenders of passenger cars are supported in much the same manner, except that the brackets are usually made of pressed steel of channel section. In order to be able to use comparatively light brackets and still

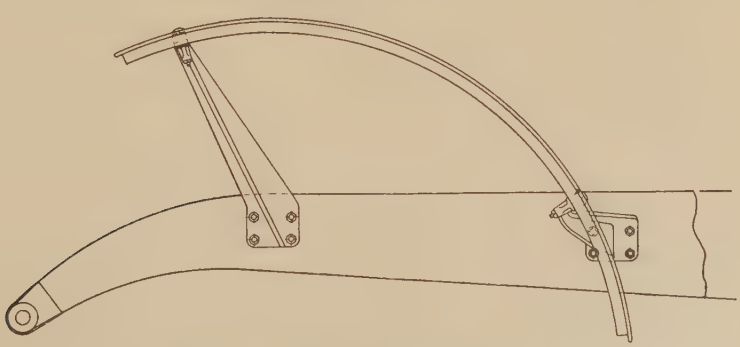


FIG. 19.—TRUCK FENDER BRACKET.

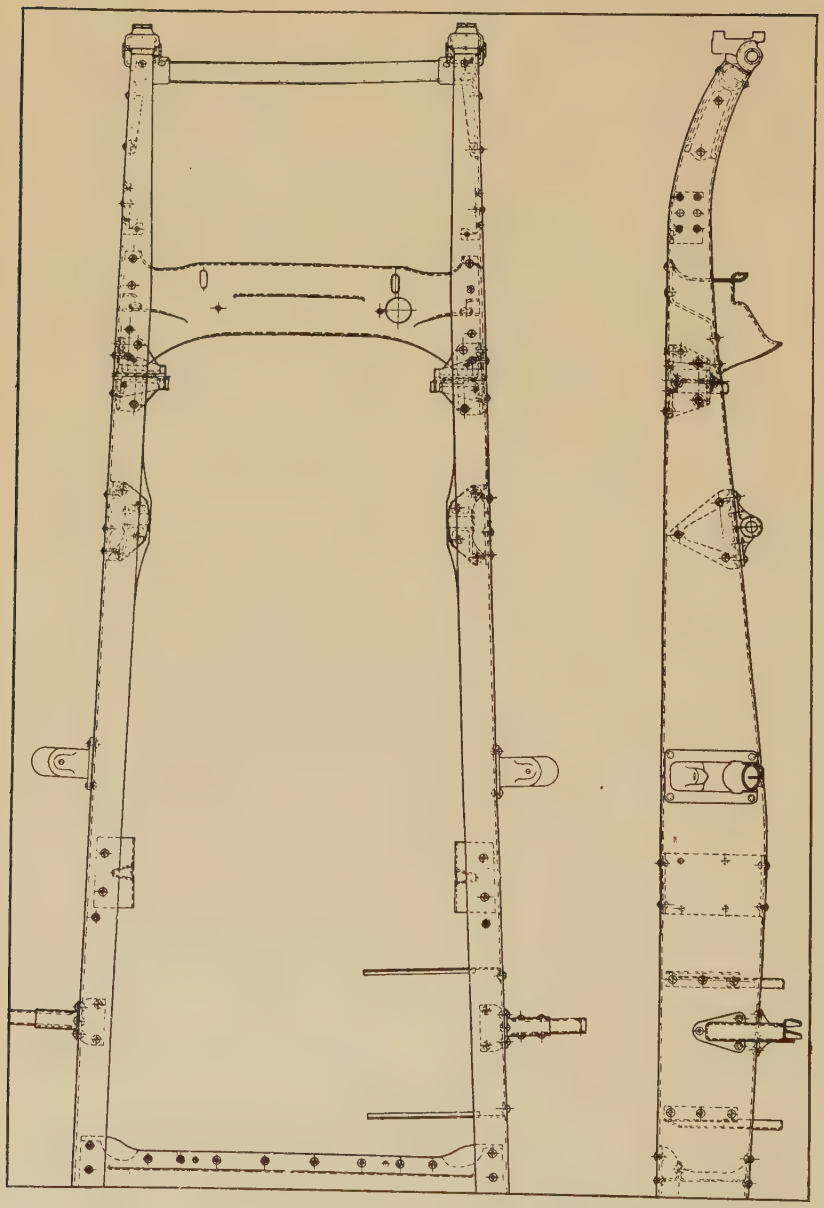


FIG. 20-A.—ASSEMBLED FRAME FOR PASSENGER CAR—FRONT END.

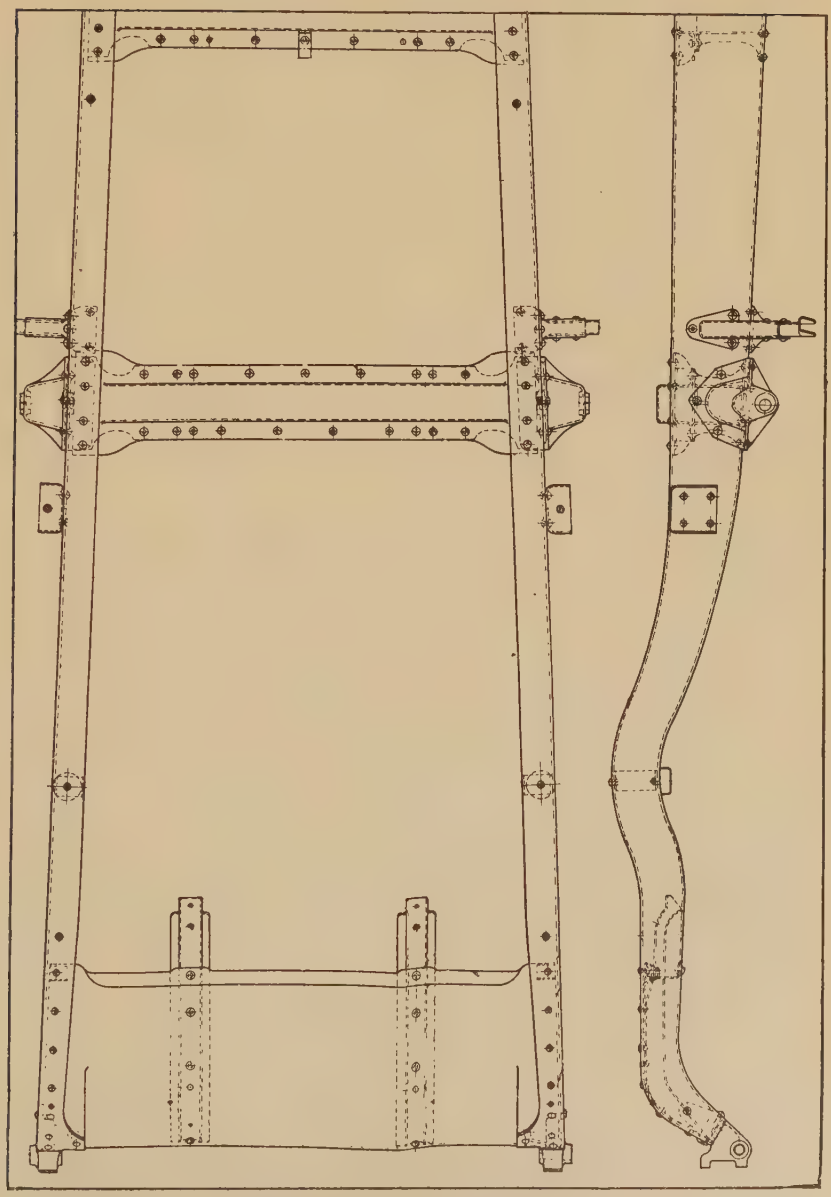


FIG. 20-B.—ASSEMBLED FRAME FOR PASSENGER CAR—REAR END.

prevent the fenders from vibrating, the brackets are often connected by a tie-rod which also supports the headlamps.

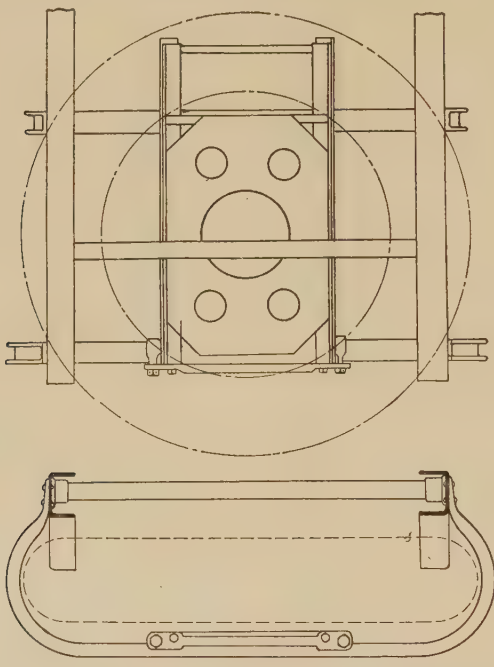


FIG. 21.—SPARE-WHEEL CARRIER FOR PNEUMATIC-TIRED TRUCKS AND BUSES.

A complete chassis frame (Studebaker) with its various brackets is shown in Figs. 20-A and 20-B.

Tire Carriers—Cars with pneumatic tires always carry a spare tire, or if demountable wheels are used, a spare wheel, and a carrier must be provided on which this spare member can be securely mounted and which permits of its easy removal. A spare tire carrier usually consists of a ring of rolled steel similar in shape to the wheel felloe on which the tire rim is mounted when in service, which is supported from a rear cross member of the frame by means of one or more brackets. Fig. 21 shows a carrier for a spare wheel with pneumatic tire as used largely on trucks and buses. The wheel is held in place by bolts passing through the same holes in the inner portion of the discs which serve for the retaining bolts when the wheel is in service.

CHAPTER XVI

Chassis Springs

The frames of all motor vehicles are supported on the axles through the intermediary of springs. Leaf springs, which are built up of a number of steel leaves or plates, are used almost exclusively, but helical springs have been used to a small extent in light vehicles of the cyclecar type, and rubber and air cushions have been proposed as a substitute for steel springs. An advantageous feature of the leaf spring is that the inter-leaf friction tends to damp out any vibration set up by road shocks, and thereby contributes toward smooth-riding qualities. However, this damping action has been found insufficient in passenger cars under modern conditions, and special damping devices, generally known under the names of shock absorbers and snubbers, are fitted in most cases.

Classification of Springs—The most widely used form of automobile spring is the half-elliptic, which is shown at A in Fig. 1. It is made up of one master leaf, each end of which is generally formed into an eye for connection to spring brackets or shackles, and a number of shorter leaves, the lengths of the leaves decreasing substantially uniformly with their distance from the master leaf, except that in springs for heavy loads the leaf or leaves nearest the master leaf sometimes extend to the end of the latter and may even enwrap the spring eyes. The various leaves are held together by a center bolt, as a rule.

For small passenger cars use is sometimes made of quarter-elliptic springs, as illustrated at L in Fig. 1. The butt end of the spring is clipped to the chassis frame, while the small end is either clipped, pivoted or shackled to the axle. All other types of spring are made up either entirely of half-elliptic springs or partly of half-elliptic and partly of quarter-elliptic members, though some springs have a scroll in addition to these elements. Scroll springs, however, have almost entirely passed out of use. At B, Fig. 1, is shown the

three-quarter elliptic spring, which consists of a quarter-elliptic top and a half-elliptic bottom member, the two being connected by a bolt at one end. At C is shown the elliptic spring, consisting of half-elliptic top and bottom members connected by bolts at both ends.

At D is shown the three-quarter elliptic scroll spring, consisting of a quarter-elliptic scroll top member and a half-elliptic bottom member joined by shackles at one end. E shows the scroll-elliptic (one end) and F the scroll-elliptic (both ends).

The spring shown at G, known as a platform spring, was formerly extensively used for high-grade cars, and for trucks. It consists of two longitudinal half-elliptics pivoted or shackled to the chassis frame at their forward end, and a transverse half-elliptic, which is shackled to the other two by double shackles or crosses. At H is shown a three-quarter-elliptic platform spring, at I an auxiliary spring consisting of a half-elliptic with plain ends. K shows the half-elliptic cantilever spring, which at one time enjoyed a considerable degree of popularity in the passenger car field, and, finally, at L is shown the quarter-elliptic cantilever spring already referred to. The half-elliptic cantilever is pivoted to the frame at or near its middle, is shackled to the frame at its forward end, and clipped, pivoted or shackled to the axle at its rear end.

Spring Materials—The common spring material, which has long been used for carriage and railway springs, is a carbon steel containing about 1 per cent of carbon. The S. A. E. specifications for this carbon spring steel are as follows:

0.95 CARBON STEEL

	Per cent
Carbon	0.90 to 1.05
Manganese	0.25 to 0.50
Phosphorus	0.04 max.
Surphur	0.05 max.

This grade of steel, if properly heat-treated, gives good results. After the spring plates have been worked into the proper shape, they are quenched in oil at about 1400° F., reheated to about 500° F. and allowed to cool slowly. These figures must not be regarded as those giving the best results, and are intended merely as indicative of the temperature ranges within which the various operations are carried out. The best quenching and drawing temperatures are usually determined experimentally in each plant, and once found,

must be closely adhered to, for any variations in temperature greatly affect the mechanical properties of the springs.

Carbon spring steels when properly heat-treated will show the following physical properties:

Tensile strength	170,000–190,000 lbs. per sq. in.
Elastic limit	140,000–150,000 lbs. per sq. in.
Elongation in 2 in.	6– 8 per cent
Reduction of area	14–18 per cent

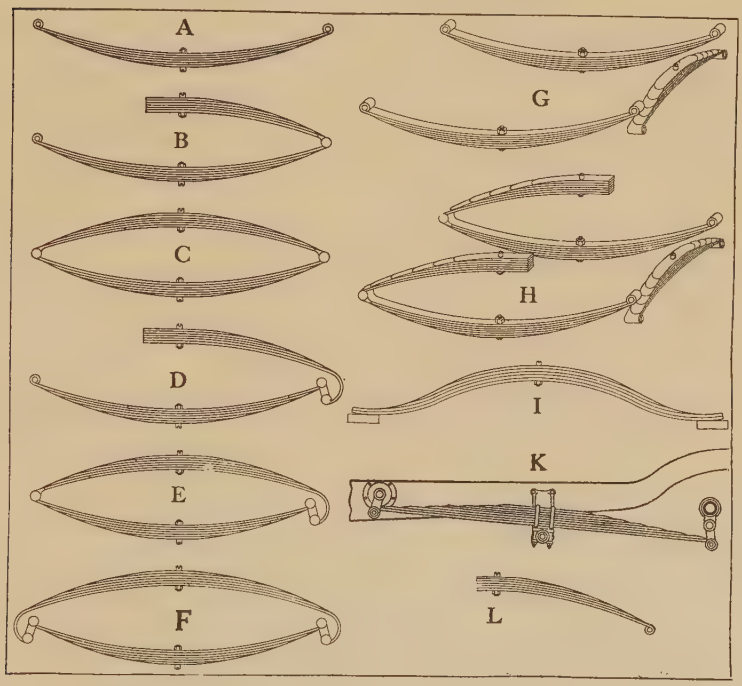


FIG. 1.—TYPES OF CHASSIS SPRING.

Chrome-vanadium spring steel made under the specifications of the Vanadium Corporation of America has the following composition:

Carbon	0.45–0.55 per cent
Manganese	0.60–0.90 per cent
Phosphorus	0.045 max.
Sulphur	0.045 max.
Chromium	0.90–1.10 per cent
Vanadium	0.15 per cent min.

For the above steel the following physical properties are claimed:

Tensile strength	205,000 lbs. per sq. in.
Elastic limit	190,000 lbs. per sq. in.
Elongation in 2 in.	10 per cent
Reduction of area	40 per cent

These test results are obtained from 7/8 in. round test specimens which are first normalized at 1560° F., held at that temperature for 15 min., then quenched in oil, and finally drawn at 900° F.

Another type of steel used in high-grade automobile springs is silico-manganese steel. The S. A. E. has standardized two silico-manganese spring steels, of the following compositions:

Steel	No. 9250	No. 9260
Carbon	0.45–0.55	0.55–0.65
Manganese	0.60–0.90	0.60–0.90
Phosphorus	0.045 max.	0.045 max.
Sulphur	0.045 max.	0.045 max.
Silicon	1.80–2.20	1.80–2.20

The heat treatment for these steels consists in heating to 1500–1650° F., quenching in oil and drawing to the hardness desired. According to A. A. Remington (Institution of Automobile Engineers), a silicon-manganese spring steel containing 0.54 C, 1.90 Si, 0.70 Mn, 0.025 S and 0.035 P showed a tensile strength of 215,000 lbs. per sq. in., an elastic limit of 184,000 lbs. per sq. in., an elongation of 6 per cent and a reduction of area of 22 per cent. Another grade containing 0.76 C, 2.17 Si, and 0.56 Mn showed the same tensile strength but a slightly higher elastic limit (190,000 lbs. per sq. in.).

Fatigue limits of Spring Steels—Endurance tests on spring steels made in England showed an average fatigue limit of plus and minus 37,600 lbs. per sq. in. for a steel containing 2 per cent of silicon, 0.92 per cent of manganese, and 0.35 per cent of carbon when quenched in water, and of plus and minus 30,650 lbs. per sq. in. for a 0.50 per cent carbon steel quenched in water. The mean value for all specimens tested was 34,300 lbs. per sq. in. The fatigue limit, it may be recalled, is the maximum stress to which a material under repetitive stresses may be subjected indefinitely without failure. It was found in the tests in England that if the specimen plates were machined, the fatigue limits were half as high again as in the rough-rolled specimens. The reason

for this is not hard to seek, for so-called fatigue failures generally start as very small cracks which widen gradually, and with a smoothly finished surface there is much less chance of such small cracks or fissures being present at the beginning.

Width and Thickness of Leaves—It has been customary for a long time to make spring plates in accordance with the Stubbs' or Birmingham wire gauge, but owing to the fact that the size increments of this gauge are not regular, the plates are made also in thicknesses varying in even thirty-seconds of an inch. In the following tables are given the sizes of the Birmingham gauge most commonly used in automobile spring practice, as well as the fractional-inch sizes, together with the squares and cubes of the thicknesses, which latter figures are convenient in making stress and deflection calculations. Sizes marked with an asterisk are included in an S. A. E. Recommended Practice.

Table I—B. W. G. Sizes of Spring Plates

No.	t	t^2	t^3
00	0.380	0.1443	0.0549
0	0.340	0.1156	0.0393
1	0.300	0.0900	0.0270
2*	0.284	0.0807	0.0229
3*	0.259	0.0670	0.0174
4*	0.238	0.0566	0.0135
5	0.220	0.0484	0.0106
6	0.203	0.0412	0.0084

Table II—Fractional-Inch-Size Spring Plates

Size	t	t^2	t^3
7/16*	0.437	0.1914	0.0836
3/8*	0.375	0.1406	0.0527
11/32	0.344	0.1181	0.0407
5/16*	0.312	0.0976	0.0304
9/32	0.281	0.0791	0.0222
1/4	0.250	0.0625	0.0156
7/32	0.219	0.0478	0.0105
3/16	0.187	0.0352	0.0065

According to the S. A. E. Recommended Practice, Nos. 2, 3, and 4 are made in widths of 1¾, 2, 2¼ and 2½ in.; the 7/16 in. size is made in widths of 3, 3½ and 4 in.; the 3/8 in. in widths of 2¼, 2½, 3, 3½ and 4 in., and the 5/16 in. in widths of 2, 2¼, 2½, 3 and 3½ in.

Theory of Leaf Springs—The simplest form of leaf spring is that containing only a single leaf. Such a spring may be considered either as two cantilever beams loaded at their ends or as a simple beam loaded at the middle. Figure 2 represents such a spring in diagram. If we consider each half of the spring as a cantilever and denote the load on one end of the spring by P, the half-length of the spring by l, the width by b, the thickness by t and the coefficient of elasticity by E, we have for the deflection of the end of the spring

$$d = \frac{1}{3}\frac{Pl^3}{EI} = \frac{4Pl^3}{Ebt^3}.$$

(See cantilever beams in any textbook on mechanics.) The bending moment at any distance x from the end of the spring is Px and the stress in the material at that point is

$$S_x = \frac{6Px}{bd^2}.$$

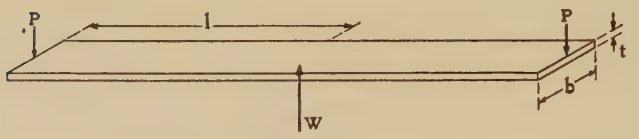


Fig. 2.

Hence, with a single leaf of uniform section over its whole length the stress due to the bending moment varies from nothing at the end to a maximum at the middle of the spring. Therefore, if a single uniform-section leaf were used, the material would be very poorly utilized, and one of the objects in using a multiple-leaf spring is to make the stress substantially uniform in all parts of the spring. Now suppose we took a number of equal leaves and assembled them as shown in Fig. 3. Then, if loads were applied to the top leaf, all of the leaves would be deflected the same amount. If there are n leaves the deflection would be the same as in the case of a single leaf subjected to a load P/n. Therefore the deflection of a spring like that shown in Fig. 3 should be

$$d = \frac{4Pl^3}{Enbt^3}.$$

As a leaf of the spring deflects there are two forces at work, viz., the force due to the load P carried, and the force

due to the internal strains. In a multiple-leaf spring, moreover, as soon as deflection occurs, there is sliding of the plates on each other, and this introduces friction as a factor. The friction between leaves is opposed to the deflection and, therefore, assists the internal forces or those due to the strains in the material. While the spring is deflecting the difference between the force due to the load on the spring and that due to the internal strain is available for overcoming the frictional resistance, and it is obvious that when this difference becomes equal to the frictional force the deflection ceases. Hence the deflection will be reduced, but the proportional reduction will be comparatively small, and will depend, moreover, on the number of leaves and on the camber of same. Experiments have shown that an eight-leaf spring in the dry condition has its vibration damped out in about three-and-one-half complete periods, or in fourteen quarter periods, each of which corresponds to one complete deflection of the spring

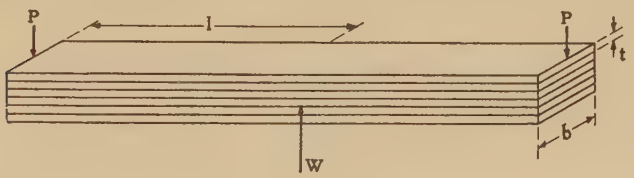


FIG. 3.

from the position of equilibrium. Hence only about 7 per cent of the energy in the spring when fully deflected is absorbed by friction during the deflection, and in a well-lubricated spring the damping action would be even materially less. This would seem to indicate that in the average passenger car front spring in average condition the loss of energy through interleaf friction during a quarter period does not exceed 5 per cent, and in an average rear spring, 8 per cent. The calculated deflection and stress should both be decreased in these proportions.

Effect of Clamped Portion—The deflection is affected also by the fact that a portion of the spring at the center is restained by the retaining clips. This part of the spring is not entirely inactive, because no matter how strong the clips or bolts may be, and how tight they may be set up, they will yield elastically under the enormous forces active at the edge of the seat. Moreover, the edge of the seat and of the clamping plate must be beveled or rounded, in order to prevent localization of stress. The clamping, however, has

some effect in reducing the maximum bending of the spring. This effect, again, is rather indeterminate, as it depends on the rigidity of the seat and of the clips or bolts, as well as upon the relative length of the seat.

Although the effective length of the spring is thus slightly reduced, it is incorrect to assume—as is sometimes done—that that portion of the spring between clips or between edges of seat is wholly inactive and must be deducted from the length. It must be remembered in this connection that even if the spring were free, it would deflect very little at a point at a distance from the center equal to half the width of the seat. For instance, if a cantilever beam 19 in. long (corresponding to a 38 in. spring) is deflected 1 in. at the free end, the deflection 2 in. from the fixed end is only about 0.010 in.

Spring Deflection—In an actual vehicle spring the leaves are of gradually decreasing lengths, and since the outer end

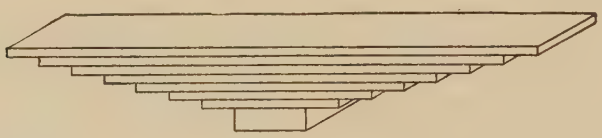


Fig. 4.

of any leaf is not supported by leaves below it, the deflection will be greater than in a spring of the form shown in Fig. 3. Reuleaux has calculated that if the lengths of the leaves decrease uniformly, as in Fig. 4, the multiplying factor will be 1.5. That is, a spring of the type shown in Fig. 4 will deflect 50 per cent more for a given load than a spring of the type shown in Fig. 3, both being of the same dimensions. However, in automobile springs the second leaf often extends out as far as the center of the spring eye, and in heavy motor truck springs, even two or three leaves support the main leaf at the eyes, in which case the multiplying factor is smaller. From the values of this factor for the extreme cases, Figs. 3 and 4, viz., 1 and 1.5, its value for any intermediate case can be approximated. For commercial springs it probably never drops below 1.25. Hence, taking 1.25 and 1.5 as the extreme values of this coefficient, we have for the deflection of vehicle leaf springs

$$\delta = \frac{1.25 \text{ (to } 1.5) \times 4Pl^3}{Enbt^3}.$$

which may be simplified to read

$$d = \frac{5 \text{ (to 6) } Pl^3}{Enbt^3} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

The larger coefficient is to be used when the length of any leaf is less than that of the preceding one by l/n; the smaller, if several of the longer leaves are substantially equal in length to the master leaf.

Equation (1) covers the case of a spring with leaves all of equal thickness. In practice springs are usually built up of leaves of different thicknesses, or, in the terms of the spring-maker, they are graded, and we may then substitute for nt^3 in the denominator of equation (1), $t_1^3 + t_2^3 + t_3^3 \ldots + t_n^3$, or Σt^3, which gives

$$d = \frac{5 \text{ (to 6) } Pl^3}{En\Sigma t^3} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

Stress Calculation—The bending moment at the middle of the spring is equal to Pl. In a spring having n leaves of equal thickness t, this bending moment is equally divided, and that on each leaf is Pl/n. Since the section modulus is $bt^2/6$, the unit stress is

$$S = \frac{\frac{Pl}{n}}{\frac{bt^2}{6}} = \frac{6Pl}{nbt^2} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

If we divide the elastic limit of the material by this stress we obtain the factor of safety. Therefore, calling the maximum safe load on the spring W ($=2P$) and the factor of safety f, we get,

$$W = \frac{Lnbt^2}{3fl} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

Dividing equation (3) by (1) we get the stress in the leaves per inch of deflection (for a spring having leaves all of the same thickness)—

$$\frac{S}{d} = \frac{\frac{6Pl}{nbt^2}}{\frac{6Pl^3}{Enbt^3}} = \frac{Et}{l^2} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

Tables III and IV, which give stresses per inch of deflection, have been calculated by this equation, a value of 28,000,000 lbs. per sq. in. being used for E.

Table III.—Stresses Per Inch of Deflection, B. W. G.-Plates

Length of Spring (Inches)	No. 00	No. 0	No. 1	No. 2	No. 3	No. 4	No. 5	No. 6
34	36,820	32,940	29,070	27,500	25,100	23,050	21,320	19,670
36	32,840	29,380	25,930	24,540	22,400	20,550	19,010	17,540
38	29,470	26,370	23,270	22,030	20,100	18,460	17,060	15,740
40	26,230	23,800	21,000	19,880	18,120	16,650	15,400	14,210
42	24,130	21,590	19,040	18,030	16,440	15,100	13,960	12,880
44	21,990	19,670	17,350	16,420	14,980	13,760	12,730	11,740
46	20,110	18,000	15,870	15,030	13,700	12,600	11,640	10,740
48	18,470	16,530	14,580	13,800	12,580	11,560	10,690	9,870
50	17,030	15,240	13,430	12,720	11,600	10,660	9,870	9,104
52	15,740	14,080	12,440	11,780	10,750	9,890	9,120	8,470
54	14,600	13,060	11,550	10,930	10,000	9,180	8,455	7,760
56	13,570	12,140	10,720	10,150	9,260	8,510	7,860	7,250
58	12,650	11,320	10,070	9,540	8,700	8,000	7,327	6,760
60	11,820	10,580	9,333	8,840	8,053	7,400	6,844	6,320
62	11,070	9,907	8,742	8,290	7,550	6,940	6,412	5,917
64	9,930	8,885	7,840	7,780	7,080	6,510	5,750	5,305

Table IV—Stresses Per Inch of Deflection—Fractional-Inch Plates

Length of Spring (Inches)	7/16	3/8	11/32	5/16	9/32	1/4	7/32	3/16
34	42,400	36,330	33,320	30,250	27,220	24,220	21,110	18,120
36	37,800	32,400	29,730	27,000	24,280	21,600	18,920	16,150
38	33,950	29,100	26,680	24,230	21,900	19,380	16,980	14,500
40	30,630	26,250	24,080	21,870	19,670	17,500	15,330	13,090
42	27,800	23,810	21,840	19,840	17,840	15,870	13,900	11,870
44	25,310	21,700	19,900	18,080	16,250	14,460	12,670	10,820
46	23,160	19,850	18,200	16,540	14,870	13,230	11,590	9,910
48	21,260	18,230	16,720	15,190	13,650	12,150	10,640	9,090
50	19,600	16,800	15,420	14,000	12,590	11,200	9,820	8,390
52	18,100	15,550	14,250	12,950	11,640	10,360	9,080	7,760
54	16,825	14,450	13,220	12,040	10,795	9,605	8,415	7,184
56	15,600	13,410	12,280	11,150	10,030	8,930	7,820	6,680
58	14,540	12,490	11,460	10,400	9,355	8,325	7,290	6,230
60	13,610	11,667	10,700	9,720	8,742	7,777	6,813	5,818
62	12,750	10,940	10,020	9,120	8,190	7,285	6,384	5,450
64	11,960	10,250	8,990	8,550	7,340	6,530	5,220	4,890

In case the leaves are of unequal thickness we can calculate the stress in each from the fact that the elastic curves of all leaves at the middle must be alike. Under these conditions the bending moment on each leaf is proportional to its moment of inertia. The bending moment on the heaviest leaf (of moment of inertia I) then will be

$$M_1 = \frac{PlI_1}{\Sigma I} = \frac{Plt_1^3}{\Sigma t^3},$$

and the unit stress in this leaf will be

$$S = \frac{\frac{Plt_1^3}{\Sigma t^3}}{bt_1^2} = \frac{6Plt_1}{b\Sigma t^3} \quad . \quad . \quad . \quad . \quad . \quad (6)$$

The maximum safe load on such a spring is:

$$W = \frac{Lb\Sigma t^2}{3fl} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (7)$$

Number and Thickness of Plates—The deflection of a cantilever loaded at the end is given by the equation

$$d = \frac{Wl^3}{3EI}$$

and the maximum stress in the material of such a lever is

$$S = \frac{Wlc}{I} = \frac{Wlt}{2I}.$$

Hence

$$S = d \times \frac{3tE}{2l^2}$$

and

$$t = \frac{2Sl^2}{3dE}.$$

Placing the coefficient of elasticity E at 28,000,000 this reduces to

$$t = \frac{Sl^2}{42{,}000{,}000\,d} \quad . \quad . \quad . \quad . \quad . \quad . \quad (8)$$

The gauge thickness closest to the result obtained should then be chosen, and if the result should come midway between the thicknesses of two gauge sizes, the length of the spring could be varied slightly and the calculation made over. After

t has been determined the necessary number of leaves may be determined by a transformation of equation (1) as follows:

$$n = \frac{5 \text{ (to } 6) \, Pl^3}{Edbt^3} \quad . \quad . \quad . \quad . \quad . \quad . \quad (9)$$

Frequency of Spring Vibrations—Every spring when supporting a given load has a definite period, or a definite rate of vibration, and the riding qualities of a car depend very much upon the period of its springs. Stiff springs have a high rate of vibration and make a car hard-riding, while soft springs have the opposite effect. As the load on the springs is increased, the rate of vibration decreases, and some cars which are hard-riding when occupied by only one or two persons, are fairly comfortable when carrying a full complement of passengers.

The two factors which affect the rate of vibration or the frequency of a spring therefore are its flexibility and the weight of the masses supported by it. In the following the relation between these factors and the frequency will be discussed.

When a chassis spring is set vibrating, as by a road shock, it is first deflected from its position of equilibrium under the load it is carrying, and a certain amount of energy is stored up in it, in the form of fiber-stress energy. Next the spring rebounds, and when it passes through the position of equilibrium the fiber stress—except that due to the static load, which we disregard here—is removed. The energy, however, has not been dissipated, but has been converted into kinetic energy, for when the spring passes through the position of equilibrium all of the masses carried by it are moving upward rapidly, and therefore possess a considerable amount of kinetic energy. Except for the small amount of energy dissipated by any damping action during the quarter cycle represented by the first half of the rebound, all of the energy originally stored up in the spring by reason of its deflection, has now been converted into kinetic energy.

After the spring has passed the position of equilibrium its upward speed decreases, hence it gives up kinetic energy, which is converted mainly into potential energy or energy of position, the masses supported by the spring being raised to a higher level against the force of gravity. When the rebound is completed, all of the kinetic energy has disappeared and has been replaced by an equivalent amount of potential energy. Next, as the spring compresses again, this potential energy is again converted into kinetic energy, and so on.

Let d = the deflection of the spring under a given impulse in feet.

R = the force in pounds required to deflect the spring 1 ft.

W = the weight supported by the spring.

n = the number of complete cycles of the spring per minute ($n/60$ per second).

g = the constant of gravity (32.16 ft. p.s.p.s.).

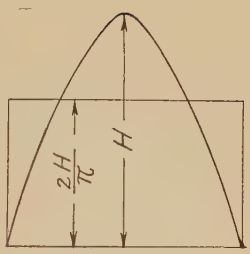


Fig. 5.

Since it takes R lbs. to deflect the spring 1 ft. and the actual deflection is d, it follows that the force of the impulse causing the deflection is dR. In deflecting the spring through a distance d, the resistance that must be overcome varies uniformly from nothing at the beginning to dR at the end of the deflection F. The energy expended (or stored up) therefore is $d\ F/2 = d^2R/2$ ft.-lbs.

Since the spring passes through n cycles per minute, the duration of one cycle is $1/n$ minute or $60/n$ second. The length of a complete cycle is equal to 4 d, because d is one-half the motion in one direction. Hence the average speed of the spring end during the cycle is

$$\frac{4d}{\frac{60}{n}} = \frac{dn}{15}.$$

The motion of the spring end is harmonic or, in other words, varies in accordance with a sine curve (Fig. 5). Now, the maximum ordinate of a sine curve is $\pi/2$ times its mean ordinate, hence it follows that the maximum speed of the spring end—the speed at which it passes through the position of equilibrium, when all of the energy carried by it is in the form of kinetic energy—is $\pi/2$ times its average speed. The maximum speed of the spring therefore is

$$v = \frac{dn}{15} \times \frac{\pi}{2} = \frac{\pi dn}{30}.$$

Now, the kinetic energy possessed by the spring-supported weight at this moment is given by the well-known equation

$$E = \frac{Wv^2}{2g} \text{ ft.-lbs.}$$

Substituting the value of v found above we get

$$E = \frac{W\pi^2 d^2 n^2}{1800\, g} \text{ ft.-lbs.}$$

As was pointed out, this is equal to the energy stored up in the spring in the form of fiber-stress at the end of the deflection, viz., $d^2R/2$.

Equating the two expressions we get

$$\frac{W\pi^2 d^2 n^2}{1800g} = \frac{d^2 R}{2},$$

from which it follows that

$$W\pi^2 n^2 = 900\, Rg.$$

and

$$n = \frac{30}{\pi}\sqrt{\frac{Rg}{W}} \text{ cycles per minute,}$$

where R is in pounds per foot of deflection. If R is given in pounds per inch of deflection, as is usually the case, the equation becomes

$$n = \sqrt{\frac{12Rg}{W}} \text{ cycles per minute.}$$

It will thus be seen that the frequency of vibration of the spring increases with its rate R and varies inversely as the weight W supported by it. It will also be noticed that the deflection d does not appear in the equation, showing that the rate of vibration is independent of the amplitude of vibration. This, however, is only because we neglect the damping action. Owing to inter-leaf friction the frequency of actual springs varies slightly with the amplitude of the vibration.

The above equation for the rate of vibration or periodicity of a spring can be presented also in a different, simpler form. The rate R of the spring is equal to the quotient of the load on it to the deflection produced by this load under static conditions, viz., W/d, and substituting this for R and inserting the values of the constants, we get

$$n = \sqrt{\frac{35{,}000}{d}},$$

where d is given in inches. This equation gives the following rates of vibration for springs with different deflections under static load:

Table V.—Relation Between Deflection Under Static Load and Rate of Vibration

Deflection	Frequency	Deflection	Frequency
2	132.5	5	84
2.5	118	5.5	80
3	108	6	76
3.5	100	6.5	73
4	94	7	71
4.5	88	7.5	68

Spring Camber—In the foregoing we have considered springs of an elementary type, with the object of simplifying the mathematics. Actual springs differ from those discussed in several ways. In the first place, conventional chassis springs are not straight when in the unloaded state, but are cambered.

Spring leaves are made from rolled stock, and after they have been cut to the right length, are bent to form arcs of circles. That is, the ordinary or conventional spring leaf, which is said to have a true sweep, is thus bent; leaves of heavy truck springs sometimes are given a double sweep, which means that their ends are curved in the opposite direction from the central portion, but this practice seems to be disappearing. In a half-elliptic spring the distance from a line joining the axes of the eyes, to the inner side of the master leaf is known as the opening of the spring, or its camber.

Formerly motor vehicle springs were made with considerable camber, the practice probably being a heritage from the horse vehicle industry, but it has been found that camber, except of small degree, is a disadvantage, and at present the tendency is to so design the springs that they are absolutely straight when under their full static load. They will then have a moderate positive camber when unloaded, and a corresponding negative camber when under their limiting shock load.

That a large camber is a disadvantage may be shown by reference to Fig. 6. The load on the spring is vertically downward, and if the spring is heavily cambered, a considerable component of the load acts in the direction of the length of the leaves, thus inducing stresses without creating any useful effects. It is the deflection of the spring which produces useful results. The load which causes deflection is equal to $P \cos a$, and this is a maximum when a is equal to zero, that is, when the spring is straight. Full elliptic springs, of course,

must be given considerable camber, as that is the only way to assure proper clearance for spring action.

Nip—When the leaves of a spring are formed, they are not all bent to the same radius, but, on the contrary—starting from the master leaf—each succeeding leaf is bent to a smaller radius than the preceding one (Fig. 7). In the terms of the springmaker's trade this is expressed by saying the leaves are nipped. When such nipped leaves are assembled and the center bolt is put through, they will be bent and thus subjected to initial stresses, which in the shorter leaves are in the same direction as that which is induced by the normal load on them, while in the longer leaves it is in the opposite direction. Thus the short leaves are subjected to a small positive initial stress by the nip, and the longer leaves to a negative stress.

The primary object of nipping undoubtedly is to assure that all leaves will remain in contact at all times, so that no

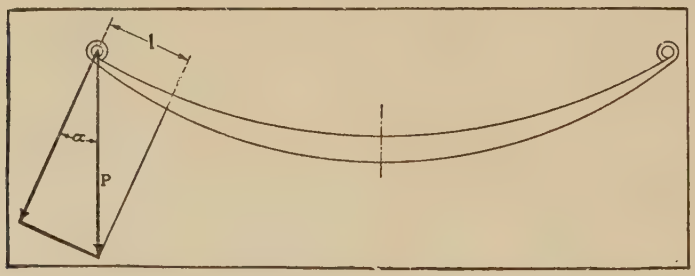


Fig. 6.—Diagram Showing Effect of Camber on Deflection.

dirt can enter between them. If the leaves were not nipped and if they were not held together by leaf clips, they would separate each time a severe rebound occurred; dust and grit would enter between them, and the problem of spring lubrication would be even more difficult than it is now.

When the load is supported by the spring, all of its leaves are stressed in the same direction, but whereas the stress in the short leaves is equal to the sum of the initial stress and that due to the load, that in the long leaves—and particularly the master leaf—is equal to the difference between that due to the load and the initial stress. The net result of nipping therefore is to decrease the stress in the master leaf and increase the stresses in the shorter leaves.

Within certain limits this is a desirable result, because it tends to make the shortest leaf the "weakest link," and it is

certainly less objectionable to have a short leaf break than the master leaf. However, large difference in the stresses on the different leaves are undesirable, because they are proof that some of the spring material is not utilized to full advantage. To prevent such large differences, the springs are "graded," that is, the shorter leaves, which are subjected to an initial positive stress, are made of thinner stock, so that a given deflection under load does not increase the stress on them as much as it increases the stress on the master leaf. Grading, moreover, is also a convenient means of obtaining fine differences in flexibility of the spring as a whole.

Since the leaves of nipped springs are pressed together with considerable force when assembled, nipping has an influence on the damping action.

Length of Springs—In the design of springs the engineer has to deal with a considerable number of factors, viz., the length and width, the thickness and number of leaves, and

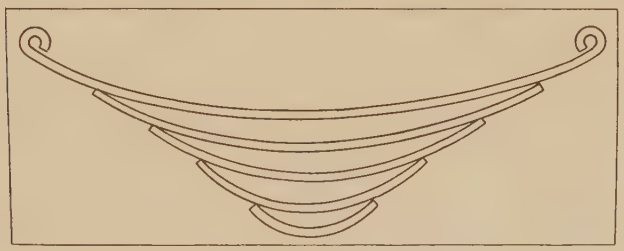


FIG. 7.—ILLUSTRATING NIP OF SPRING PLATES.

the elastic limit of the material. Of these the first two generally are determined by empirical rules based upon experience. Longer springs make for easier riding, because the greater the length, the more the spring can be allowed to deflect without danger of breakage. The stress in the material remains the same if the length varies as the square of the thickness, the load and all other dimensions remaining the same, in which case the permissible deflection varies as l^2/t or as $l^{1.5}$.

Half-elliptic front springs of passenger cars vary in length from 30 to 34 per cent of the length of wheelbase, the percentage usually being lowest in cars with the longest wheelbase. It may be explained in this connection that the length of the spring is measured between the centers of the eyes, and this, of course, varies with the load; it is advisable in referring to the length to define it as the free length or the

loaded length, which latter means the length under rated load. Front springs for passenger cars are practically limited to three widths, viz., 1¾, 2 and 2¼ in.

The length of passenger car rear springs (half-elliptic) varies from 43 to 48 per cent of the length of wheelbase, and here, too, the percentage is usually the lowest in the longest cars. Rear springs cover four widths, namely, 1¾, 2, 2¼ and 2½ in., and the largest widths naturally go together with the greatest lengths.

Lengths of motor-coach springs average as follows:

Table VI.—Lengths of Coach Springs

Wheelbase (Ins.)	Front Springs (Ins.)	Rear Springs (Ins.)
Up to 190	38–40	54–58
190–200	40–43	58–60
200–220	43–46	60–62
Over 220	46–48	62–64

Widths of motor-coach front springs vary from 2¼ to 3½ in. and of rear springs, from 2½ to 4 in.

Average lengths of motor truck half-elliptic springs are given in Table VII, on next page.

Flexibility—The most important characteristic of a chassis spring is its deflection under its rated static load. The deflection of front springs is purposely kept small, because a large deflection of these springs makes the steering erratic, while, on the other hand, the cost of spring material increases directly as the deflection under static load, provided the factor of safety is kept constant, and, of course, front-spring flexibility has little bearing on passenger comfort.

In passenger cars the deflection of the front springs under full static load usually lies between 1¼ and 2 in. Rear springs of passenger cars are probably the most important factor in connection with passenger comfort, and they are therefore made long and flexible. The deflection of these springs under full static load varies between 4 and 7 in., being greatest in the larger and more luxurious cars.

Factors of Safety—It has been found that the ratio of maximum stress under shock to the static load stress is greater in the case of front springs than in that of rear springs, which is probably due (at least in part) to the fact that most of the load on the front springs (the power plant) is

Table VII.—Motor Truck Springs

FRONT SPRINGS, HALF-ELLIPTIC

Load Capacity, Tons	Length (Inches)	Width, (Inches)
¾	38 to 40	2
1	38 to 40	2¼
1½	40 to 42	2½
2	42 to 44	2½
3	44 to 46	2½ to 3
4	46 to 48	3
5	48 to 50	3

REAR SPRINGS, HALF-ELLIPTIC

Load Capacity, Tons	Length, Inches	Width, Inches
¾	48 to 52	2
1	48 to 52	2¼
1½	50 to 53	2½
2	50 to 53	2½
3	52 to 54	3
4	52 to 55	3
5	54 to 56	3½

dead load, whereas a good deal of the load on the rear springs is live load, which is cushioned, moreover, by the upholstery. The factor of safety, based upon the elastic limit and the full static load, should be made from 3.25 to 3.5 in the case of front springs, and from 2.25 to 2.5 in the case of rear springs.

In order that the springs may be able to protect the car frame and body from road shock, adequate clearance must be allowed between the frame and axles. This clearance should be slightly greater than the total deflection of the spring under dead load. If the clearance is thus limited the spring can never be stressed to more than a little over twice its normal stress, so that danger of breakage is almost eliminated if the elastic limit is from 2.5 to 3.5 times the normal stress.

Allowance for Stresses Due to Braking—Where the springs are required to resist the braking torque in addition to carrying the load on them, a reduction must be made in the stress in the spring material under static load, in order that there may be sufficient reserve strength to withstand the

braking stresses. The allowance which should be made can be judged from the following considerations. Since we have called the load on each half of a semi-elliptic spring P, the load on one wheel is substantially $2P$. Now assume that the wheel is locked by the brake and is sliding on a dry, hard surface, which will have a coefficient of friction of about 0.6. Then the drag on the wheel rim is $1.2P$, and if the wheel radius is r, this produces a moment $1.2Pr$ which tends to twist the axle around its wheel spindle axis. This moment is balanced by a couple due to reactions on the spring at its eyes. The total length of the spring is $2l$ and if we denote the reaction at each eye by R, the couple is equal to $2Rl$.

At the rear end of the spring the reaction is downward, in the same direction as the load on the spring, and the stresses induced in the spring material by the braking torque therefore are in the same direction and add to those induced by the load. On the contrary, the reaction on the spring at the

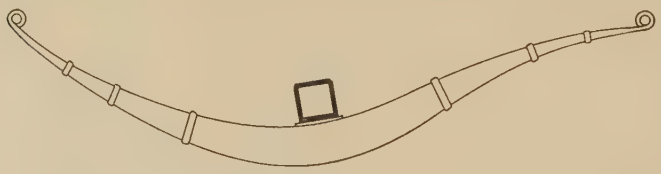


FIG. 8.—FORM ASSUMED BY CHASSIS SPRING UNDER BRAKING TORQUE.

forward end is upward and therefore opposed in direction to the load at this end of the spring, for which reason the forward part of the spring is relieved of stresses by the braking torque. Under the action of braking torque the spring assumes a shape somewhat as shown in Fig. 8.

Equating the moment due to the brake torque with the wheel sliding on the ground, to the couple due to the reactions on the spring eyes, we get

$$2Rl = 1.2\,Pr.$$

from which it follows that

$$R = 0.6\left(\frac{r}{l}\right)P.$$

That is to say, the reaction on each spring eye due to braking is directly proportional to the load on the spring and varies with the ratio of the wheel radius to the length of the spring.

It might appear that the safe working stress under static

load should be reduced in the proportion of $(R + P)/P$, but this is not correct, for the reason that R is the maximum reaction under braking torque which can possibly occur, whereas P is only the static load, the effect of which may be doubled by shock. It is therefore more logical to reduce the stress under static load in the proportion $(R + 2P)/2P$. Inserting the value of R in terms of P, this becomes

$$\frac{[2P + 0.6\ (r/l)\ P]}{2P}$$

For front wheels and springs the ratio r/l is usually about 0.75 and for rear wheels and springs, 0.50. Inserting these values in the foregoing equation, we find that to allow for braking stresses the safe stress under static load should be reduced in the proportion of 1.225 to 1 (nearly 20 per cent) in the case of front springs, and 1.15 to 1 (13 per cent) in the case of rear springs.

Sometimes, in order to provide for the stresses due to the braking torque, an extra half leaf is included in the rear half of the spring, on top of the master leaf. It is secured at the center by the clips which hold the spring to the axle, while near the outer end and at intermediate points it is tied to the other leaves by leaf clips.

Application of Formulae and Data—There is no difficulty in applying the equations developed in the foregoing to a spring having leaves all of the same thickness, as it is only necessary to select the length and width in accordance with the rules given, and then solve for the necessary number and thickness of leaves required by means of equations (8) and (9). If the result of the solution for the number of leaves involves a fraction close to 0.50, the length can be varied slightly and the calculation carried through once more.

The problem is a rather more involved one, however, when—as is usually the case—the leaves are to have nip and to be graded. We will explain the procedure in that case with the aid of an example. Let it be desired to design front springs for a moderate-sized passenger car of, say, 115-in. wheelbase. Let the maximum weight on each front spring be 800 lbs. According to the rule that the length of front springs varies between 30 per cent and 34 per cent of the length of wheelbase, the range from which to select is 34½ to 39 in. We will choose 38 in. Since this is a medium weight car, the intermediate of the three widths for passenger car front springs, 2 in., will be suitable. The deflection of the spring under full static load may be assumed to be 1¾ in.

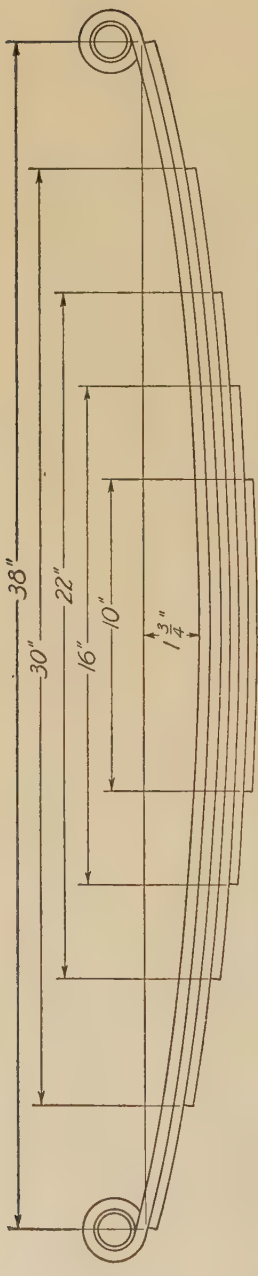


FIG. 9.—FRONT SPRING FOR WHICH CALCULATIONS ARE CARRIED THROUGH.

If we use alloy steel of 180,000 lbs. per sq. in. elastic limit and allow a factor of safety of 3.5, the maximum safe stress under static load will be $180{,}000/3.5 = 51{,}500$ lbs. per sq in. This, however, must be reduced by 20 per cent to allow for braking stresses, which makes the maximum safe stress under static load 41,200 lbs. per sq. in. In the case of the master leaf there will be a negative initial stress of the order of 10,000 lbs. per sq. in., and we can add this to the maximum safe stress under static load, making it 51,200 lbs. per sq. in. This stress will correspond to a deflection of 1¾ in., hence the stress per inch of deflection will be $51{,}200\ 1\frac{3}{4} = 29{,}300$ lbs. per sq. in.

By now referring to Table III we see that for a 38-in. spring the stress per inch of deflection is very nearly equal to this value in a No. 00 plate. Consequently, we choose a No. 00 plate for the master leaf.

To determine the other leaves required we make use of equation (2) which can be transformed to read

$$\Sigma t^3 = \frac{5.5Pl^3}{Ebd}$$

(the mean value 5.5 being used for the constant). Inserting values we get

$$\Sigma t^3 = \frac{5.5 \times 400 \times 19^3}{28{,}000{,}000 \times 2 \times 1\frac{3}{4}}$$

$$= 0.1542.$$

In selecting the other leaves we must be guided by the consideration that the shorter leaves will have a positive stress induced by the nip, hence they must be of such thickness (or thinness) that the

stress induced in them by the static load is less than the permissible safe stress of 41,200 lbs. per sq. in. (or 23,500 lbs. per sq. in. per inch of deflection). With the aid of Table I we make up the following selection:

One No. 00 (master leaf)	$t^3 = 0.0549$
One No. 1	$t^3 = 0.0270$
One No. 2	$t^3 = 0.0229$
Two No. 3 ($t^3 = 0.0174$)	0.0348
One No. 4	$t^3 = 0.0135$
	$t^3 = 0.1531$

which is very close to the 0.1542 calculated as required. This combination of plates will give the required deflection of 1¾ in. for a load P of 400 lbs., and it is only necessary to determine what must be the initial shapes of individual leaves so the nip stresses will have the proper values.

It has been found that lengthening the first and second supporting leaves below the master leaf slightly beyond their theoretically correct lengths tends to equalize stresses throughout the master leaf and therefore, has a beneficial effect. We will assume that the first leaf below the master leaf extends as far out as the centers of the spring eyes. Now, we found that deflecting the master leaf 1¾ in. induces a stress of 51,500 lbs. per sq. in. in it, hence to produce a negative initial stress of 10,000 lbs. per sq. in. it must be deflected 1¾/5.15 = 0.34 or about 11/32 in.

We now make a drawing of the spring in the form we wish it to have when assembled, in which we are guided by the consideration that it is desirable to have it straight under full static load, so that the "opening" of the free or unloaded spring will be equal to 1¾ in. plus one-half the bore of the spring eye. Upon this drawing we superpose another one of the master leaf in its free condition, making the ends of the master leaf in the two conditions coincide, but having the centers separated by 11/32 in. An arc of a circle through the three points mentioned determines the form of the master leaf in the free state.

Now as regards the leaf which directly supports the master leaf, from Table III we find that a plate No. 1, 38 in. long, when deflected 1 in. will be subjected to a stress of 23,270 lbs. per sq. in., and hence when deflected 1¾ in., to 40,700 lbs. per sq. in. We want it to be stressed to 41,200 lbs. per sq. in., which is almost the same, hence this leaf can be made

to the form which it is intended to have in the assembled spring.

The third leaf, a No. 2 plate, will be subjected to a stress of $1\frac{3}{4} \times 22{,}030 = 38{,}600$ lbs. per sq. in., and therefore must be given a nip such that it has an initial positive stress of $41{,}200 - 38{,}600 = 2600$ lbs. per sq. in. To determine the nip necessary to produce this initial stress in this leaf we make use of the equation for stress in a cantilever beam loaded at the end, viz.,

$$S = \frac{3Etd}{2l^2},$$

from which it follows that

$$d = \frac{2Sl^2}{3Et}.$$

The half-length of the third leaf is 15 in. Remembering that the thickness is 0.284 in. and the stress desired 2600 lbs. per sq. in., inserting values we get

$$\frac{2 \times 2600 \times 15^2}{3 \times 28{,}000{,}000 \times 0.284} = 0.049 \text{ in.}$$

The fourth leaf, a No. 3 plate, will be stressed to $1\frac{3}{4} \times 20{,}100 = 35{,}150$ lbs. per sq. in., hence it may be given a nip stress of $41{,}200 - 35{,}150 = 6050$ lbs. per sq. in. This leaf has a half-length of 11 in., hence it must be nipped

$$\frac{2 \times 6050 \times 11^2}{3 \times 28{,}000{,}000 \times 0.259} = 0.067 \text{ in.}$$

The next leaf is of the same size, No. 3, but it is only 8 in. long on each side of the center, hence its nip should be

$$\frac{2 \times 6050 \times 8^2}{3 \times 28{,}000{,}000 \times 0.259} = 0.036 \text{ in.}$$

Finally, the shortest leaf, a No. 4 plate, will be subjected to a stress of $1\frac{3}{4} \times 18{,}460 = 32{,}360$ lbs. per sq. in. and therefore can have a nip stress of $41{,}200 - 32{,}360 = 8840$ lbs. per sq. in. Its length to each side of the center is 5 in., hence the nip can be

$$\frac{2 \times 8840 \times 5^2}{3 \times 28{,}000{,}000 \times 0.238} = 0.022 \text{ in.}$$

The above calculated nips may be used without exceeding the safe working stress in any of the plates. There is another thing to be considered, however, namely, the pressures required to produce the nip stresses, for the pressures taking up positive and negative nips will automatically balance, and this is almost certain to change the nip stresses. It is therefore necessary to calculate the pressures on the plates (by the center bolt) necessary to take up the various nips. For this we use the cantilever equation

$$P = \frac{Ebt^3d}{4l^3}.$$

We must apply this equation to the master leaf, which requires a downward pressure, and to the four short leaves, which require upward pressures, and then see whether the sum of the upward pressures required by the short leaves is equal to the pressure required by the master leaf. Since the terms E, b and 4 are the same in all equations we can simplify the calculations by omitting them and using the expression t^3d/l^3. Applying this to the master leaf we get 0.00000271, while the sum of the results obtained for the four shorter leaves is 0.00000503. Hence it requires nearly twice the pressure to press the short leaves upward into position than to press the master leaf down, and in order to get the correct initial or nip stress in the master leaf, since nip stresses and nips are proportional, we should reduce the calculated nip of each of the shorter leaves in the proportion of 503 to 271 or by 46 per cent. This makes the various nips as follows: Third leaf, 0.0265 in.; fourth leaf, 0.036 in.; fifth leaf, 0.020 in. and sixth leaf, 0.012 in.

With these nips the shorter leaves will be stressed somewhat less than 41,200 lbs. per sq. in. under static load, but this, of course, is no disadvantage. These nips are quite small, and nips are larger, of course, in the case of a rear spring with about three times the initial deflection.

Eccentrated Springs—Most semi-elliptic springs extend equal distances from each side of their support, but occasionally front springs are made with one end shorter than the other and are then said to be eccentrated. Three-quarter elliptic springs are eccentrated as a rule, for if both ends of the half-elliptic member were made of the same length, the rear part of the spring, which includes the quarter-elliptic member, would tend to deflect twice as much as the forward part under a certain load increment, which would result in non-uniform loading of different parts of the spring. With

an eccentrated half-elliptic spring the reactions at the two spring eyes will be to each other inversely as their respective distances from the center of the spring seat and since the stress in the spring is proportional to the length, stress distribution in such a string will be uniform. Figure 10 shows an eccentrated half-elliptic spring, the lengths l_1 and l_2 of the two ends being unequal.

Cantilever Springs—The equations developed in the earlier part of this chapter apply strictly only to half-elliptic and quarter-elliptic springs and must be modified when applied to other types. In Fig. 11 is shown a diagram of a half-elliptic cantilever spring supporting a load W. The reaction of the axle on the rear end of the spring evidently is equal to W, while the reaction of the frame on the spring at the forward end is

FIG. 10.—ECCENTRATED HALF-ELLIPTIC SPRING.

$$W_1 = \left(\frac{l_2}{l_1}\right) W,$$

and the pressure of the spring on its center support is

$$W + W_1 = W\left(\frac{l_1 + l_2}{l_1}\right).$$

The deflection of the rear end of the spring is

$$d_r = \frac{5.5 W l_2^3}{Enbt^3}$$

while that of the forward end is

$$d_f = \frac{5.5 W\ (l_2/l_1) l_1^3}{Ebnt^3}.$$

However, the forward end of the spring cannot move relative to the frame, and spring action of the forward end results

in a rocking motion of the spring as a whole around its center pivot, which increases the deflection of the rear end. Since the rear end is longer than the forward end in the proportion l_2/l_1, the deflection of the rear end due to the spring action of the forward end will be multiplied in this proportion and therefore will be

$$\frac{5.5W\ (l_2/l_1)^2 l_1^3}{Ebnt^3}$$

Adding this to the deflection of the rear end due to its own spring action we get for the total deflection

$$D = \frac{5.5W}{Ebnt^3}\left[l_2^3 + \left(\frac{l_2}{l_1}\right)^2 l_1^3\right].$$

By making $l_1 = l_2$, that is, for the case of a half-elliptic

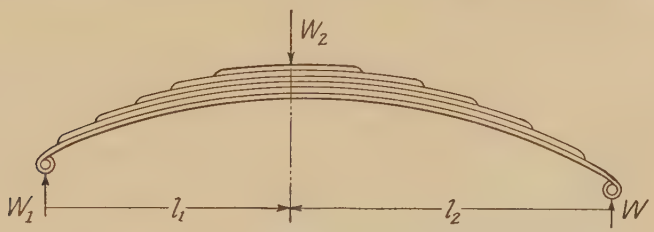


FIG. 11.—DIAGRAM OF HALF-ELLIPTIC CANTILEVER SPRING.

cantilever spring with both ends equal, we get

$$D = \frac{5.5W}{Ebnt^3}(2l_1^3).$$

Since W is the entire load carried by the spring and is therefore equal to $2P$, P being the load at one end of a half-elliptic spring for the same total load, it follows (by comparing with equation 1) that the deflection of a half-elliptic spring of given dimensions is four times as great when used as a cantilever than when used as a regular half-elliptic. This explains why half-elliptic cantilever springs are always shorter and contain more leaves than regular half-elliptics for the same load and deflection.

Center Bolt—The separate leaves of a leaf spring are usually held together by means of a center bolt. Center-bolt dimensions have been standardized by the S. A. E. The form

of the standardized bolt is as shown in Fig. 12 and the standardized dimensions are given in Table VIII.

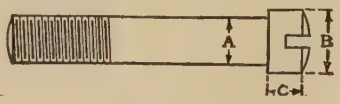


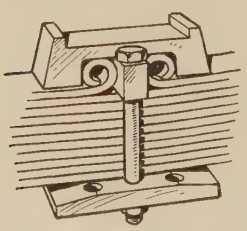
Fig. 12.—Spring Center Bolt.

Table VIII.—S. A. E. Standard Center Bolts

Spring Width, Inches	A	B	C
1¼, 1½, 1¾ and 2.	5/16	½	3/8
2¼, 2½ and 3.....	3/8	9/16	3/8
3½ and 4.....	7/16	5/8	½
Over 4.....	½	¾	½

All threads S. A. E. Standard. U. S. Standard cold-pressed blanks, tapped with S. A. E. Standard thread, shall be used for spring center-bolt hexagon nuts.

An objectionable feature of the center bolt is that it, or the hole which is required for it, materially weakens the spring, and breakages through the center-bolt hole are of rather common occurrence. For this reason many attempts have been made to eliminate the need for a center bolt. Figure 13 shows a construction due to Jonas Woodhead & Sons of Leeds, England. The master leaf is divided at the center and each of the adjacent ends is provided with an eye. In the design shown, the pressure plate forms a bridge over the eyes, while a yoke and clamp bolts hold the other leaves to the bottom plate. The usual U-bolts pass over the pressure



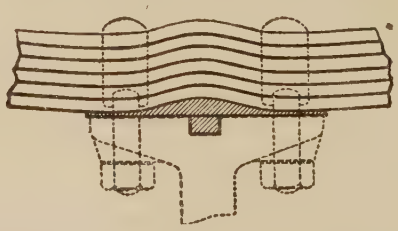


Fig. 13 (left).—Woodhead Spring with Master Leaf Made in Halves.

Fig. 14 (right).—Center Hump of Tuthill Spring.

plate, through the bottom plate and through the spring seat on the axle. In another form, bolts pass through the eyes at adjacent ends of the master leaf and through the side walls of a sort of spring box, which in turn is secured to the spring seat on the axle by means of studs integral with the side plates of the box.

Another scheme patented in England by a manufacturer of commercial vehicles (Karrier Motors, Ltd.) consists in forming the leaves with lateral projections at their middle, on both sides, these projections entering slots formed in the side walls of a spring box, which, together with its cover or pressure plate, is secured to the spring seat of the axle by bolts. Forging lateral projections on the leaves must be rather expensive, however.

The Tuthill Spring Co. of Chicago, in order to obviate the need for a center-bolt hole, gives the plates a hump at the center, as shown in Fig. 14. This construction does not hold the leaves together when the spring is not assembled on the vehicle, but that function is now usually satisfactorily performed by spring clips.

Spring Seats and Clips—Half-elliptic springs are secured to the spring seats of the axles by means of studs, box-clips or U-clips. These holding-down members are preferably made of alloy steel with a low carbon content, which has high ductility and therefore will absorb a large amount of work before it breaks. The shank of the clip is usually made of a diameter equal to one-fourth the width of the spring, and is cut with an S. A. E. thread. Hexagon nuts are used on these clips, of extra length (about twice standard), and they are locked either by heavy-type spring lock washers or by lock nut and cotter pin. The distance between clips is made as small as the design of the spring seat permits, in order to keep down the restraint on the spring action of the leaves. A pad of some soft material must be placed between the spring and its seat, to prevent concentration of stress, and good results are obtained with two layers of 8-oz. duck soaked in white lead. In the case of front springs a thin wedge is usually placed between spring and seat in addition, to give the caster of the front wheels that may be desired.

Figure 15 shows a box-clip in place. Some engineers have shown a preference for a somewhat different design, in which the upper portion of the clip is flattened only very little on the inside, or not at all. This undoubtedly reduces the strain at the corners of the clip, and it may also reduce the restraint on the spring action of the plates.

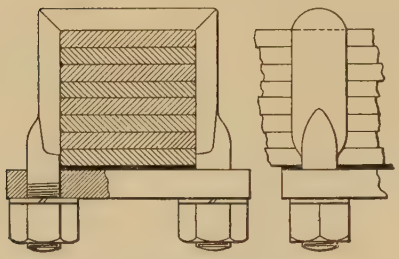


FIG. 15.—BOX CLIP.

Rear spring seats are either made rigid with the axle housing, or free to turn thereon. The former arrangement is used in connection with the Hotchkiss drive, in which case all torque reaction on the rear axle has to be taken up on the springs; while the latter arrangement is used where a torque arm or torque tube is provided. Owing to the fact that chassis frames of passenger cars taper from front to rear, the springs, as a rule, are not at right angles to the axle, and the spring seats must be arranged accordingly. Some springs are mounted on top of the axle, the front axle being dropped and the frame "kicked up" over the rear axle to secure the clearance necessary for spring action. But in many cases—especially if the springs are located directly below the side rails and have a comparatively large range of deflection—they are mounted below the axle or underslung (Fig. 16).

If the rear axle is of the built-up type the spring seat

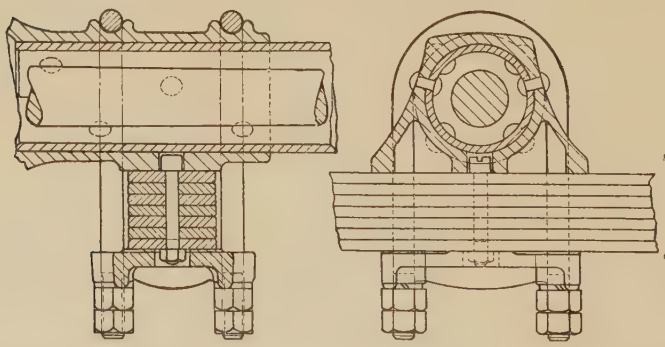


FIG. 16.—SPRING-SEAT FOR UNDERSLUNG SPRING, RIVETED TO AXLE TUBE.

is often cast integral with the brake supporting bracket or spider. In the case of axles with pressed-steel housing the seat often surrounds the axle only half way and is secured to it by welding, the spring being secured to the axle by means of U-clips which pass over the axle tube and through holes in a clamp plate or spring block. A rear axle spring seat welded to the axle tube is shown in Fig. 17, while Fig. 18 shows a rear axle spring seat that rocks thereon, both being designed to take a spring held in place by U-clips.

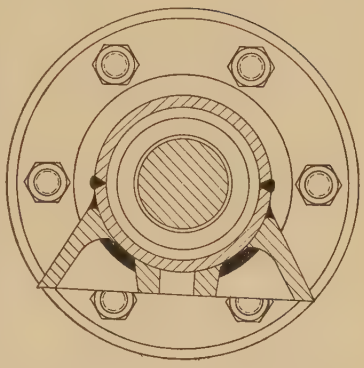


FIG. 17.—SPRING SEAT WELDED TO AXLE TUBE.

Front and rear solid axles usually have spring seats forged integral with them. However, most of the axles of this type used on motor trucks are manufactured by specialists who supply a number of truck makers, and as the spring centers on trucks of nearly the same maximum weight often differ considerably, it is necessary either to make the seats so long (in the direction of length of the axle) as to cover the whole range of spring center distances, or else provide a spring seat separate from the axle that can be moved along same as required and held in place by a dowel. The former method results in a rather awkward appearance, and axle manufacturers, as a rule, seem to prefer the adjustable seat.

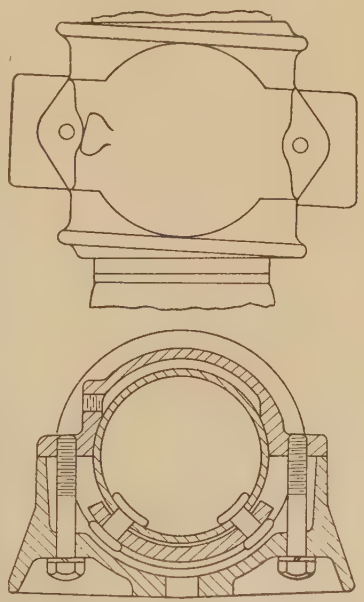


FIG. 18.—SPRING SEAT FREE TO ROCK ON AXLE TUBE.

Pressure Block—Instead of having a box-clip bear directly on the spring leaves, a pressure block is frequently placed between it and the top leaf or master leaf. The one shown

in Fig. 19 is made with grooves for the clips, with a hole or socket for the head of the center bolt, and with a curved bearing surface, which latter is claimed to obviate localization of stress at the end of the spring seat and to render the whole length of the spring available for elastic deflection. In this connection it is of interest to mention a pressure block developed by John I. Thornycroft & Co. in England. It has curved strips running across its under side at the hold-down bolts, so that theoretically there is line contact between the pressure block and the master plate. The spring seat, moreover, has the same formation, and the claim made for this design is that it eliminates restraint of the spring curvature.

Rebound Clips and Reverse Leaves—When a wheel strikes a road obstacle the spring near it is compressed and stores up energy. Immediately after the compression has come to an end, the spring distends again, and if the blow to the wheel was a heavy one, the rebound will carry the chassis

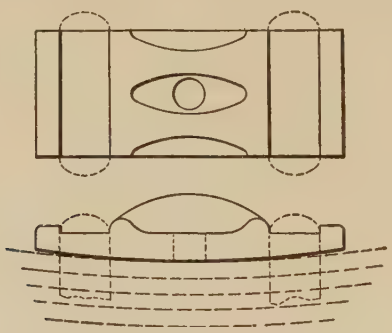


FIG. 19.—PRESSURE BLOCK.

far beyond its position of equilibrium. The master leaf will thereby be curved in the reverse direction, and as it is not supported by the other leaves in this direction, it is easily stressed beyond the elastic limit.

There are several methods of preventing injury to the master leaf from this cause. The most common consists in the use of rebound clips by which part of the load is transferred during sharp rebounds from the master leaf to the second and third leaves. As shown in Fig. 20, the rebound clip is preferably riveted to the end of the shortest leaf which it surrounds, and a tubular spacer is slipped over the bolt to prevent the leaves being clamped between the ends of the U-shaped clip, whereby their free play under ordinary

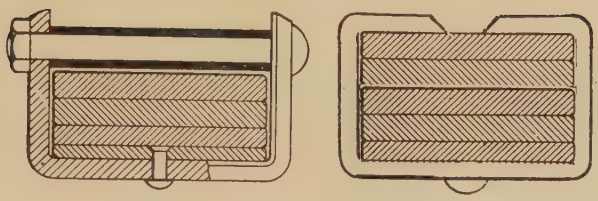


FIG. 20 (LEFT).—REBOUND CLIP.
FIG. 21 (RIGHT).—CLINCHER-TYPE OF REBOUND CLIP.

running conditions would be hampered. There is also another, simpler form of clip, known as the clinch clip, which is simply a piece of flat steel bent into rectangular form, with the joint at the middle of one of the long sides. This form of clip, as illustrated in Fig. 21, is used mainly near the spring eyes, which hold it in position.

Another method of limiting the rebound consists in placing a couple of leaves with reverse camber on top of the master leaf, as illustrated in Fig. 22. These reinforce the master leaf during rebound, and prevent its breakage. Their effect on the deflection of the spring must, of course, be considered in the calculation thereof. Finally, most high-speed cars now are provided with spring dampers (shock absorbers), which reduce the rebound and quickly damp out oscillations of the spring-suspended system.

Alignment—The leaves of a spring with rebound clips are held in alignment by the latter, but in low-speed vehicles it is sometimes undesirable to use rebound clips and some other means must then be provided for keeping the leaves in line. One plan is to raise a central longitudinal rib on top of the leaves for a certain distance from the end, the rib on one leaf entering a corresponding gutter on the next. This method is quite successful, but unfortunately it does not permit of the use of clips on the leaves. Another plan consists in providing the ends of the leaves with lips, by drawing out the leaf stock laterally in the forge and bending the lips at right angles, as shown in Fig. 23. A third method consists in slotting the end of the leaf longitudinally and raising a nib on the leaf below it. Since the majority of

FIG. 22.—SPRING WITH REVERSE OR PRELOADING LEAVES.

FIG. 23.—SPRING LIPS.

motor vehicle springs now carry rebound clips, these alternate methods of assuring alignment are only rarely met with.

Spring Points—In order that the spring material may be utilized most efficiently, the stress should be uniform throughout. If the leaves were arched to a uniform radius of curvature over their whole length and the ends cut off blunt, that portion at the end which projects beyond the supporting leaf directly below would not be stressed as highly as the rest of the leaf, and, besides, there would be a point of concentration of stress where the supporting leaf ends. To prevent this, spring leaves formerly were tapered over the projecting portion, and their ends were shaped into "points" that were attractive to the eye. A collection of such spring "points" is shown in Fig. 24, but they are really of historic interest only. While springs with tapered and pointed ends are nice to look at, they are expensive to produce, and in modern automobiles it is practically impossible to see any part of the springs unless one crawls under the car, hence money spent on them merely for the sake of improved appearance is wasted.

At present the great majority of springs are made without a taper on their thickness. In order to gradually reduce their stiffness at the ends, they may be tapered down in width to one-half or one-third their normal width (referred to as a blunt diamond leaf point) or practically to nothing, a prac-

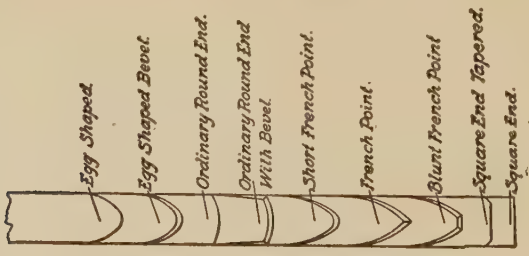


FIG. 24.—SPRING "POINTS."

(Only the two at the right and one similar to the third from the right are now in practical use.)

tice followed to a considerable extent in England. In the United States it has become customary to cut the ends off blunt straight across, and the projecting portion of the leaf is given such a form that it bears on the leaf above over its whole length only under maximum deflection.

Spring Eyes, Bolts and Shackles—The spring eyes may be either turned in or out, or they may be in line with the master leaf, as illustrated at A, B, and C in Fig. 25, respectively. In-turned eyes are best, as they are easier to make than the central eyes, and there is less danger of their opening up under pressure than there is in the case of out-turned eyes. According to S. A. E. specifications, the spring ends must be finished to a width 1/16 in. less than the nominal width of the spring, with plus and minus tolerances of 0.005 in. in the case of passenger-car, and 0.010 in. in the

FIG. 25.—SPRING EYES.

case of truck springs. These dimensions must be maintained far enough along the length of the spring to permit free motion of the shackle, or free sliding motion in the case of flat-end springs.

Spring eyes generally are bushed with phosphor bronze, and spring bolts for use with such bushings are hardened and ground. The nominal wall thickness of spring-eye bushings of all sizes has been standardized at 1/8 in., and tolerances have been established on the bores of both bushed and unbushed spring eyes. Steel tubing also has been used for bushing spring eyes, a piece of seamless tubing of the proper length being pressed into the eye and then reamed out, but this practice is less popular. The object of the bushing, of course, is to make it possible to renew a wearing surface at low cost.

Truck springs occasionally are provided with elongated eyes known as box-eyes, which slide on rollers on the spring bolts; they are also made without eyes (flat ends) at either one or both ends, the ends then sliding on combined wear plates and guides.

Shackles are made of drop forgings and the two for each spring end are usually separate and of the form shown in

Fig. 26, but sometimes they are connected by a cross-bar as illustrated in Fig. 27. A typical spring bolt is shown in Fig. 28. Spring bolts must be secured against rotation in one or both shackles, so that all relative motion will be between them and the spring eye or its bushing. The bolt shown in Fig. 28 is formed with serrations below its head. The whole bolt is hardened, and when it is forced through the hole in the soft-steel shackle, the serrations dig into the surface of the hole and the bolt is firmly fixed in the shackle. Another method of fixing the bolt consists in providing the eye of the shackle with a boss which is drilled for a clamp bolt that passes slightly below the surface of the spring bolt. The eye is split and is therefore rigidly clamped down on the spring bolt. Still another method consists in milling off the seat for the bolt head on the shackle with a shoulder against which one of the faces of the hexagon head (or a flat on a round head) abuts.

Spring bolts are drilled part-way through their center and radially at the middle of their length, and the end of the central hole is tapped for a lubricating fitting. Where the radial hole emerges at the surface, either a flat or an oil groove is cut, to give the lubricant a chance to spread over the surface of the bearing.

The distance apart of spring brackets should be set so that the shackles will be vertical when the spring carries its normal static load, whether they be in compression or tension. The chief reason for this is that the load on the shackle-bolt bearing is then a minimum. With some types of spring, to prevent reversal due to excessive rebound, shackles of the non-reversible type, as illustrated in Fig. 29, are used.

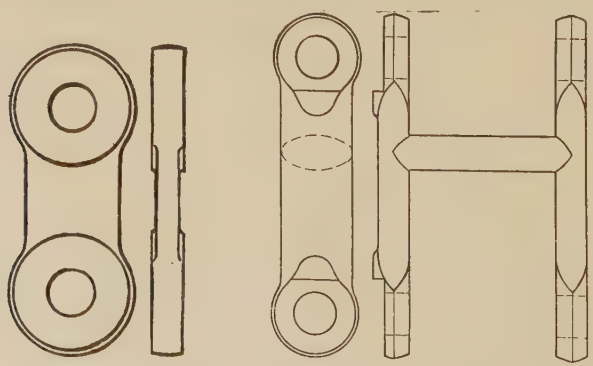


Fig. 26 (left).—Ordinary Spring Shackle.
Fig. 27 (right).—Spring Shackles with Cross Bar.

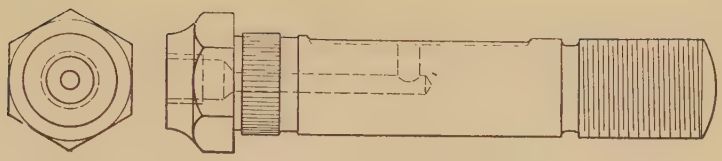


FIG. 28.—SHACKLE BOLT.

A patented spring shackle that is widely used particularly on low-priced cars is the Tryon, illustrated in Fig. 30. Hollow pins are pressed into unbushed spring eyes and are provided with tapered ends which have bearings in the shackles. The two shackles are drawn together by a central bolt having a flat spring under its head. Rocking motion takes place only at the tapered bearing surfaces, and since the hollow pins form large reservoirs for lubricant, which are replenished from time to time through the lubricator fittings on the outer shackle, these surfaces are always well lubricated, while any wear taking place on them is taken up automatically through a small range by the flat spring, after which it can be compensated for by adjustment of the nut on the central bolt.

Ball-bearing shackles also have come into use. The object of the ball bearings in this case is not to prevent friction but to minimize wear and thus keep the shackles silent for an indefinite period of time.

Springs with Flat Bearing Surfaces—Instead of the master leaf being formed with eyes, its ends may be left flat and made to bear against so-called wear plates. Such bearings, of course, are self-adjusting for wear; and, moreover, it is possible to provide larger bearing areas with them than with shackle pins, and thus to reduce the rate of wear. A development of this scheme is made use of in Buessing trucks and buses, and is illustrated in Fig. 31. The master leaf, which is supported by the two leaves adjacent to it over its entire length, is formed with a cup or hemisphere at each end, this hemisphere being lodged in a socket formed in a shoe, which latter slides over a wear plate on the spring bracket. A yoke spanning the end of the spring prevents

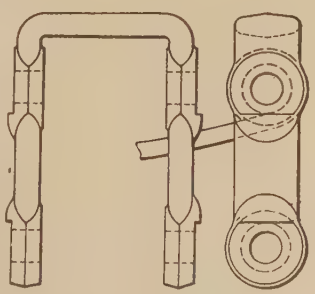


FIG. 29.—NON-REVERSIBLE SHACKLE.

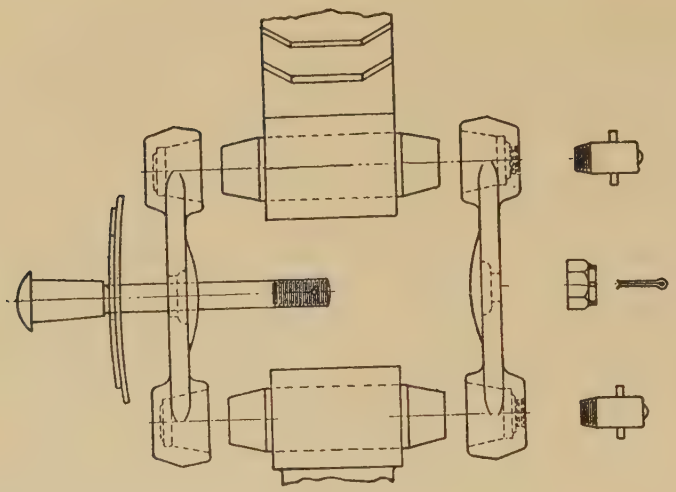


FIG. 30.—TRYON SELF-ADJUSTABLE SHACKLE.

the hemisphere from leaving the socket during severe rebounds.

This construction is used on a six-wheeled chassis, the general arrangement of the rear springs of which is as shown in Fig. 32. Two half-elliptic springs are used, one on each side. Below each spring there is a spacer bar which is pivoted to lugs underneath the rear axle housings. The two spacer bars are held in alignment by a system of diagonal bracing as shown, and the spring seats are pivotally mounted on the central tubular cross-bar, straddling the spacer bar. Another spring with flat ends is illustrated in Fig. 38 farther on, and is there referred to in connection with another feature.

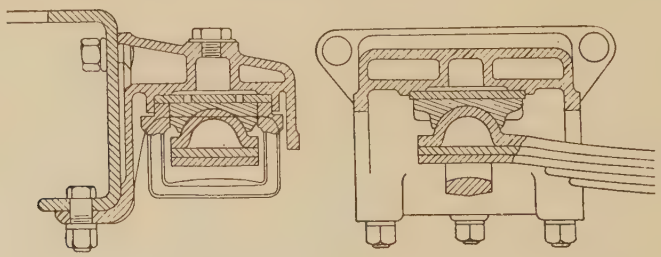


FIG. 31.—SPRING WITH SLIDING SHOES AND WEAR PLATES.

Rubber Spring Anchorage—Instead of metallic shackles, which require constant attention in the way of lubrication, and usually are one of the first parts of the car to develop appreciable wear and become noisy, rubber spring anchorages or shock insulators, an invention of A. F. Masury, chief engineer of International Motors Co., have come into extensive use. As illustrated in Fig. 33, there is no spring eye, but instead the two main leaves are bent over at right angles and are lodged in a cavity in a molded rubber block, which in turn is compressed in a housing riveted to the frame and

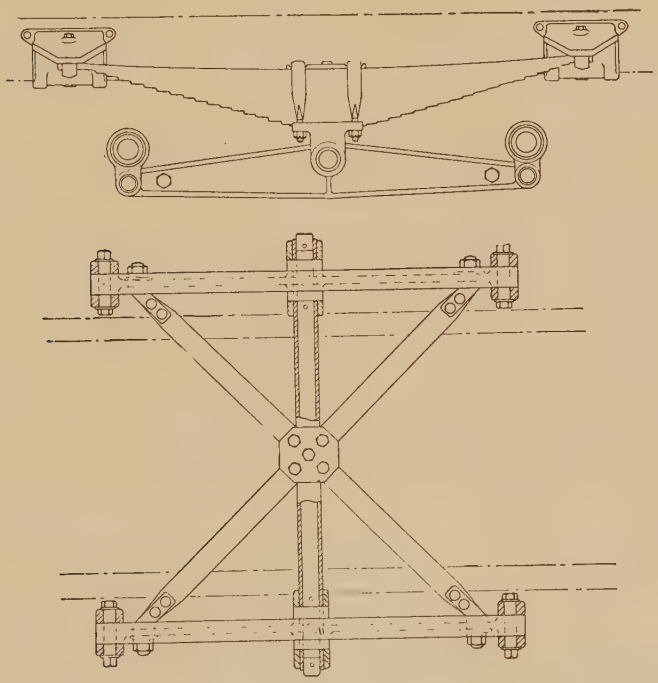


FIG. 32.—REAR SUSPENSION FOR SIX-WHEELED CHASSIS.

replacing the spring bracket. A bolted cap holds the assembly of spring end, rubber block and, possibly, a pair of pressed steel cups fitting projections on the rubber block, in position under a predetermined compression. These rubber spring anchorages require no attention, are permanently noiseless and have the further advantage that they do not become dirty with oil, and therefore facilitate cleaning of the vehicle. The

rubber blocks, of course, also are flexible and add to the flexibility of the suspension system, hence the name "shock insulator." Prof. Becker of Charlottenburg Technical College, who made tests on a Yellow Coach Co. bus fitted with these rubber spring anchorages, found that their rate was about 2800 lbs. per inch, increasing somewhat with the load.

Elastic bushings for shackles have been introduced in Europe under the name Silentbloc. These consist of two metal sleeves, one within the other, separated by a layer of non-metallic material of great elasticity and strong adherence to metal (probably a rubber compound). This material is forced into the space separating the two metal sleeves) in a layer of from 2 to 5 mm. thickness, and will permit one of the sleeves to be rotated up to 60° with respect to the other,

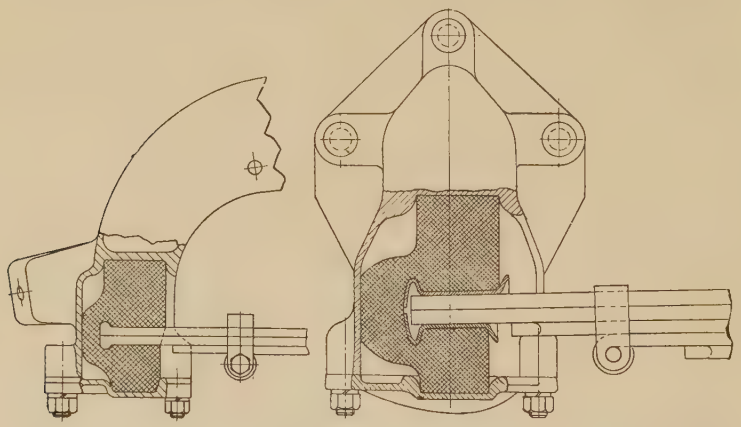


Fig. 33.—Rubber Shock Insulator for Front and Rear Springs.

after which the sleeves will return to their original relative angular position if the torsion is removed. Similar rubber-insulated bushings in which the rubber is vulcanized to the metal surfaces are made in this country.

Spring shackles of rubberized fabric are made by the Bellflex Corp., Detroit, in two forms, as illustrated in Fig. 34. The first is designed to take the place of the ordinary metallic shackle, over which it has the advantage that it does not require lubrication and remains silent indefinitely. The second is for use where driving thrust must be transmitted. In each case two thicknesses of the rubberized fabric are used, which are secured by bolts to the end of the master

leaf and to a spring bracket. Metal washers of suitable form are interposed between bolt-head, fabric, and metal backing, to distribute the stress in the fabric as much as possible.

Rubber Spring Bumpers—To limit the deflection of springs (and, consequently, the stresses to which they may be exposed) without having hard steel parts abut directly, use is often made of rubber spring bumpers. In the case of front springs they are generally placed directly on top of the spring, and held in place by a pressed-steel retainer, which is made with flaps that are clamped under the spring clips. One such bumper is shown in Fig. 35. In the case of the rear axle, since the spring in most cases is not directly below the frame side rail, the bumper is mounted on the axle tube

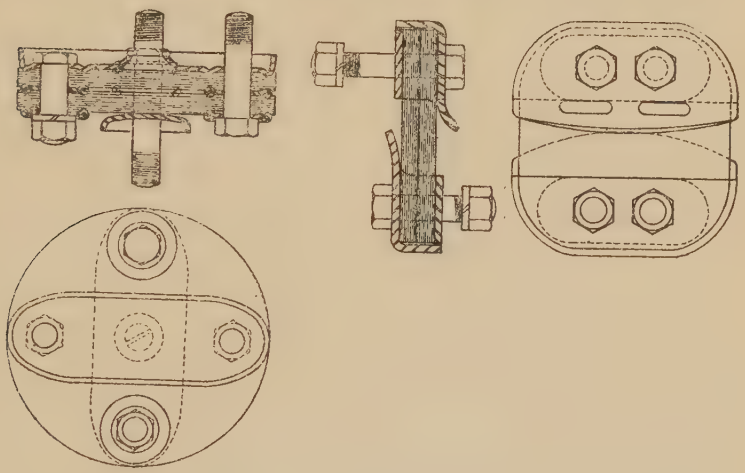


FIG. 34.—BELFLEX RUBBERIZED FABRIC SHACKLES

in the proper place under the side rail. Alternately, the bumper may be fastened to the side rail on its under side, and this arrangement is often more convenient.

Rubber spring bumpers may be regarded as auxiliary springs of a high and variable rate, and some work with the object of developing a rubber auxiliary spring for motor buses has been done by the London General Omnibus Company, the results having been given in an article by Dr. F. W. Lanchester in *The Automobile Engineer* (London) for June, 1928. An egg-shaped rubber spring was used, as illustrated in Fig. 36, and its effect on the suspension system is brought

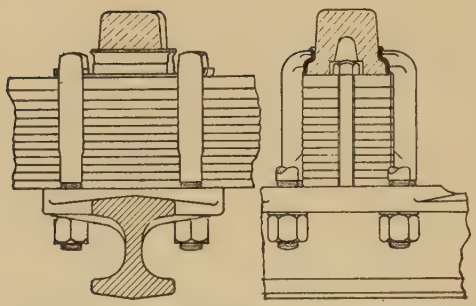


FIG. 35.—METHOD OF MOUNTING RUBBER BUMPER ON FRONT SPRING.

out by the load-deflection curves in Fig. 37. A rubber spring of this type naturally has a certain damping action, since rubber when deformed always absorbs work and generates heat internally. Referring to the possible future of springs of this type, Dr. Lanchester says they can be designed to take the place, functionally, of the shock absorber, providing the necessary damping; at the same time they will diminish the quantity of steel necessary in the laminated springs, and, further, by ultimately acting as a buffer, they will prevent excessive spring stress.

Variable-Rate Springs.—The difficulty of securing good riding qualities under all conditions with vehicles in which the loads on the spring vary within wide limits has been referred to already, and much effort has been spent on designs of spring in which the rate, that is, the pounds per inch of deflection, instead of being constant as in the conventional spring, increases with the load. One such design, used on Armleder trucks, is illustrated in Fig. 38. The spring is made with flat ends which bear against wear plates of rounded form. When the load is small, contact occurs at or near the outer edges of the wear plates, hence the effective length

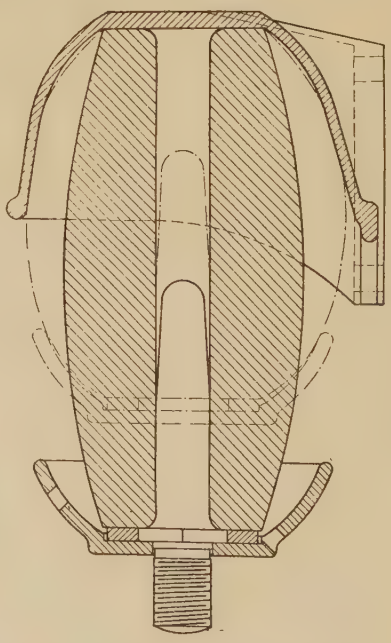


FIG. 36.—RUBBER AUXILIARY SPRING FOR MOTOR BUS.

of the spring is comparatively large, and the spring is soft. As the load increases the points of contact move toward the inner edges of the wear plates, and the springs become much stiffer. It is evident that the load-deflection curve can be varied within considerable limits by suitably shaping the outline of the wear plates.

A German firm (Buessing) uses a design in which, as the spring flattens out, a greater length at the middle is rigidly supported, so that the effective length also becomes smaller, while Marmon has used a hardwood "cam" between the halves of an X-type transverse spring, which renders more or

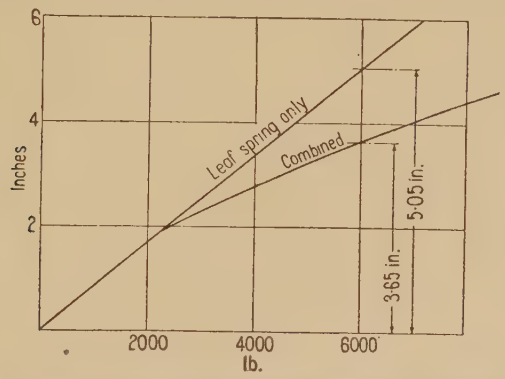


Fig. 37.—Deflection Diagram of Steel-and-Rubber-Spring Combination.

less of the center portion of the spring ineffective by supporting it rigidly.

Helper Springs—Helper springs or auxiliary springs are much used on vehicles which must carry widely varying loads. These springs, which may be either of half-elliptic or quarter-elliptic type, may be secured to either the axle or the frame. When the load on the springs reaches a certain value the free end of the helper spring abuts against a wear plate on the other part, that is, the axle if it is secured to the frame, and vice versa. From that point on the two springs carry the load jointly and their rates are added. For instance, if the main spring has a rate of 400 lbs. per in. and the auxiliary spring of 600 lbs., then the rate of the combination is 1000 lbs. per in. A quarter-elliptic helper spring secured to the frame side rail and bearing down on wear plate formed integral with the spring seat on the rear axle is shown in Fig. 39. Half-elliptic helper springs, which are

mounted above the main half-elliptic springs, are sometimes so made that their halves come into action for different total loads, by placing the wear plates at different levels, for instance, which gives three different rates for the combination. Motor buses generally are provided with helper springs in order to make them easy riding both when completely loaded and when nearly empty. One well-known bus which carries 1725 lbs. on each rear spring when empty and 2535 when fully loaded, has a main rear spring with a rate of 425 lbs. per in. and a helper spring which makes the combined rate 700 lbs. per in.

Interleaf Friction—Tests of interleaf friction under various conditions were made in the laboratory of the Detroit Steel Products Co. and the results were given in an S. A. E. paper by Tore Franzen. It was found that the static coefficient of dry friction was 0.35, and the coefficient of sliding friction, dry, 0.22. When the plates were lubricated the

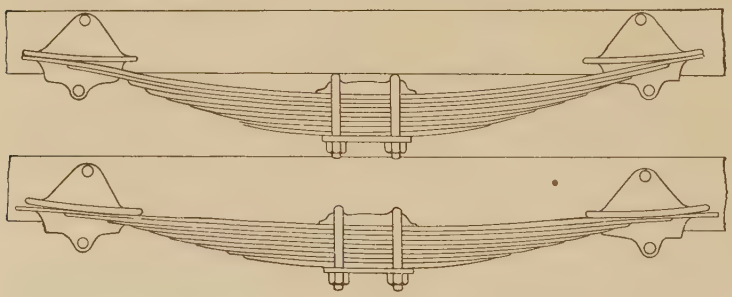


FIG. 38.—VARIABLE-RATE SPRING.

coefficient of friction increased with the load. This is contrary to the behavior of the coefficient of friction in well-lubricated bearings, where it is inversely proportional to the load, but the difference is due probably to the comparatively rough surfaces of the spring plates. The coefficient was lowest when the leaves were lubricated with a mixture made up of equal volumes of cylinder oil and powdered graphite. The coefficient of sliding friction with this lubricant varied only between 0.04 and 0.06 throughout the practical range, and the coefficient of static friction was only slightly higher, being the same as the coefficient of sliding friction within the lower load range and reaching 0.08 at the upper end of the load range. Under lubrication with a light oil, the coefficient of sliding friction rose from about 0.04 at low loads

to 0.10 at high loads, and the coefficient of static friction varied between 0.20 and 0.28 throughout nearly the whole range. With the excess oil wiped off the coefficient of sliding friction was about 0.18 through most of the range, and the coefficient of static friction, about 0.36.

The total friction between leaves is nearly proportional to the number of leaves, and the damping action increases rapidly with the number of leaves.

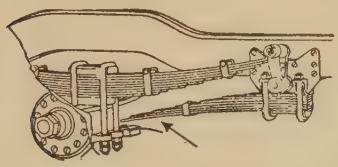


FIG. 39.—HELPER SPRING.

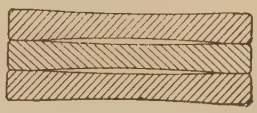


FIG. 40.—CONCAVE PLATES.

Concave Spring Plates—Although friction between leaves is desirable to a certain extent, because it damps rebound and oscillations, the leaves are lubricated where they bear against each other. To enable the leaves to retain lubricant, they are sometimes rolled with a concave-surface section so that a grease-retaining space is formed between adjacent ones (Fig. 40). Lubrication of leaves is necessary chiefly because without it the springs would become dry and squeak unpleasantly, and eventually they would rust, and their spring action would be much impaired. Good lubrication makes the action of the springs more nearly uniform, and to obtain it, springs are now often fitted with covers or gaiters of leather or artificial leather.

Spring Finish—Complete springs are furnished in three conditions or states of appearance. When all plates are left in the condition in which they come through the heat-treating process, the spring is said to be black. If the outside surface of the eyes and the upper exposed portion of the plates is ground or otherwise cleaned of scale, the spring is said to be half-bright. Finally, if all exposed surfaces of the plates except those on the under side are ground or polished, the spring is said to be bright.

CHAPTER XVII

Independent Suspensions

In the conventional motor vehicle, two road wheels are carried at opposite ends of a rigid axle, and when one of the wheels passes over an obstacle, the whole system is affected; in fact, every part of the assembly turns around the point of ground contact of the other wheel as a fulcrum. This means that the unsprung weight which is set in motion on passing over the obstacle is comparatively large, and—as was explained in a previous chapter—large unsprung weight generally is a cause of poor riding qualities. It is, moreover, objectionable also on other grounds besides that of riding comfort. When a wheel hits an obstruction, the force of impact is directly proportional to the unsprung weight carried by it. The impact causes strains in the wheels, axles, springs, etc., and it also has a destructive effect on the road. Hence, it is highly desirable to reduce the unsprung weight as much as possible.

It is chiefly for the above reason that much work has been done in recent years, particularly in Europe, on methods of suspending each wheel independently. In the case of the front wheels, independent suspension offers the further advantage that when the wheels are not connected by a rigid axle, they do not form a single vibrating system, and the possibility of front-end vibration is therefore eliminated, or at least greatly reduced.

One criterion of the merits of a system of independent suspension therefore is the proportion in which it reduces the unsprung weight affected when one wheel hits an obstruction. Of course, the system must be judged also on other grounds, such as simplicity, strength and reliability, cost of construction, and general appearance.

The systems of independent suspension which have been developed so far may be conveniently divided into four classes, as follows:

1. Vertically slidable stub axles.
2. Stub axles secured to the ends of transverse leaf springs.
3. Driving axles with axle tubes capable of a vertical rocking motion relative to the center housing.
4. Cranked stub axles fulcrumed on the chassis frame.

Vertically Slidable Stub Axles—The first system may be easily visualized as follows: Imagine a chassis frame and body, the two forming a rigid assembly. At each corner of the frame there is a bracket which comprises a vertical cylinder, in which is located the vertical part of a stub axle of L or T form, floating between coiled springs. The brackets at the front of the chassis really take the place of the axle ends or steering heads, while the stub axles are the steering knuckles and are provided with knuckle arms for connection to the steering gear. In the case of the rear brackets means must be provided to maintain the axle spindles at right angles to the axis of the car.

A car of this type has no axles, properly speaking. The chassis springs are interposed between the stub axles and the brackets, and the latter therefore are spring-supported. At the driving end, the differential and driving-gear housing is mounted on the frame, hence is spring-supported, and axle shafts with universal joints in them extend from it to the stub axles. The springs generally employed with this form of suspension are helical steel springs, and since these have no damping action, it is necessary to provide shock absorbers, which are generally built right into the wheel mounting. In one design, by the French engineer Sensaud Lavaud, lozenge-shaped rubber blocks, piled on top of one another, are used. One thing that may be urged against this form of suspension is that it is rather difficult to effectively lubricate the sliding surfaces and protect them against dust, water and splashing mud.

Lancia Front-Wheel Suspension—The best example of an independent wheel suspension of the vertical-sliding-motion type is that at the forward end of the Lancia Lambda, which is illustrated in Fig. 1. This car was first exhibited at the Paris automobile show in 1922 and has been in production for seven years. In this design the steering head is supported from the car frame and radiator shell by tubular trusses. One tube at both top and bottom lies in the vertical plane through the front-wheel axes, and an additional tube at both top and bottom runs toward the rear, the lower one of the pair being fastened to the car frame. This tubular truss

construction evidently assures great rigidity with little weight. A long tubular knuckle pin is used, which is capable of both a rotary and a sliding motion in the steering head.

The weight of the forward end of the car is supported on the steering knuckle through a coiled spring surrounding the tubular knuckle pin, while at the lower end of this pin

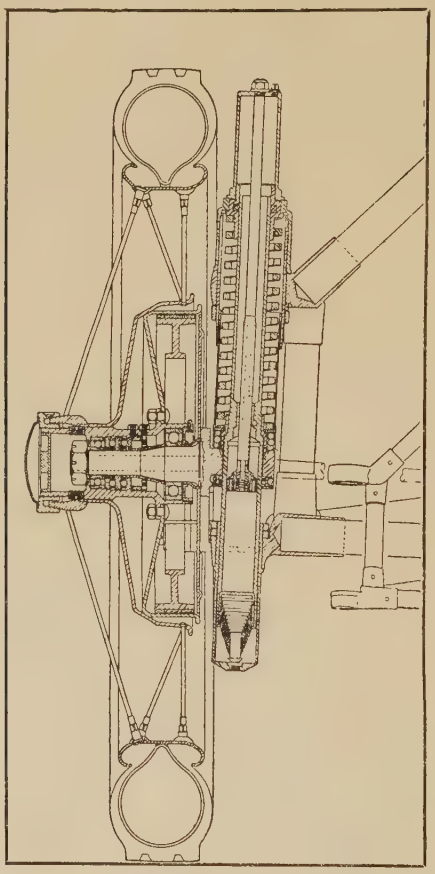


Fig. 1.—Lancia Lambda Front-Wheel Suspension.

there is a volute-type of steel spring which cushions the rebound. An hydraulic shock absorber is incorporated in the design. Within the lower part of the hollow knuckle pin there is a stationary piston, and as the knuckle pin moves up

and down with the spring action, the space between this piston and the neck in the pin in which the piston rod has a bearing, is varied, and oil is forced through a central hole in the piston rod from the space referred to, to a space in the upper part of the knuckle pin, and vice versa. The piston rod is anchored in a cap over the steering head.

Suspension by Cross-Springs—Many experimental cars have been built, and a few cars have gone into production, in which rigid axles are replaced by transverse leaf springs. These cross-springs are generally of the quarter-elliptic type, their butt end being fastened to a base mounted on the frame, which in the case of a driving axle is the driving-gear housing. Either three or four such springs extend from the center housing or base toward each side, so that the wheel support may be held firmly against both horizontal and vertical forces. If four springs are used, two are placed above and two below the axle shaft, two being in front and two to the rear of the shaft. It is possible, however, to replace the springs on one level by rigid links, and only one vertical tier of springs is needed if radius rods run from the wheel supports to the frame.

In a construction of this type, most of the weight of the springs themselves moves up and down with the chassis frame, and unusually good riding qualities can be obtained. It has been objected to it that the wheel tread varies with the play of the springs, and that this is bound to cause sideward slip of the tire tread on the ground, and consequent tire wear. However, by careful design this effect can be minimized, and the author does not believe that this needs to be a serious factor, especially with balloon tires, which have considerable lateral flexibility and are capable of compensating for moderate changes in width of tread.

A suspension of this type is used on the French Cottin-Desgouttes car, the rear construction of which is shown in Fig. 2. There are four quarter-elliptic springs on each side of the center housing, bolted to spring seats cast on the housing. At its outer end the master leaf of each spring is clipped to a separate spring eye. These spring eyes are connected by means of spring bolts to the wheel support, which is in the form of a bearing shell. Within it there are two ball bearings, these supporting a short driving shaft to which the hub of the driving wheel is keyed. The axle shafts both consist of three parts, one carrying one of the differential side gears, another carrying one driving wheel, and the third intermediate between the first two and connecting to each of them through a universal joint. It will be noticed that the

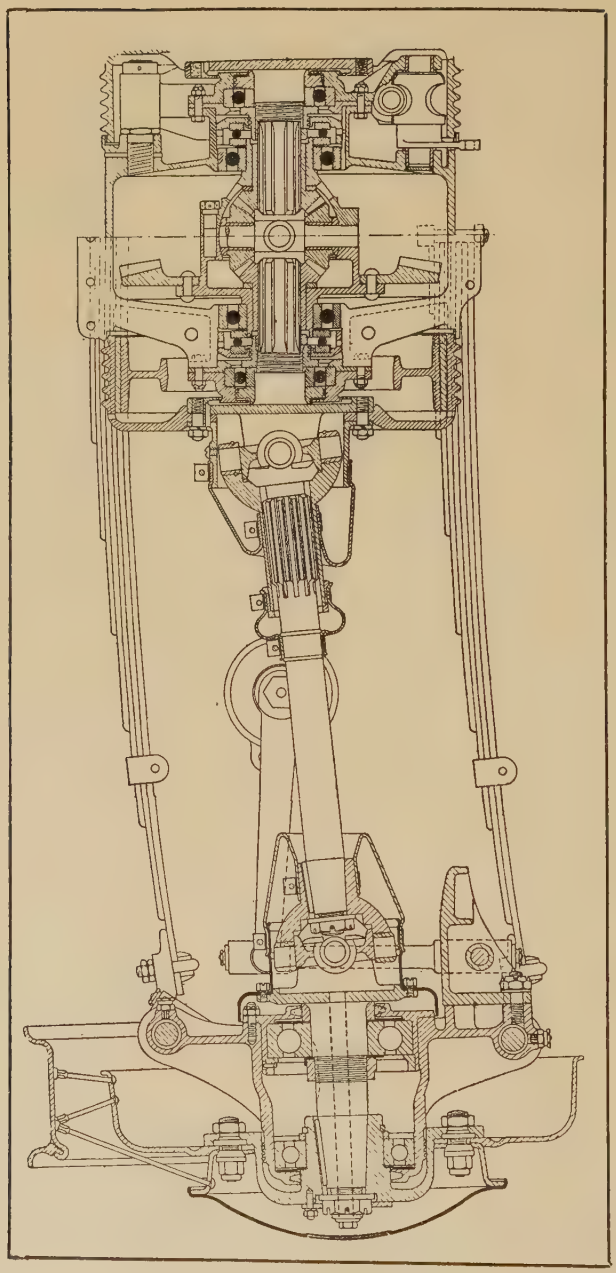


FIG. 2.—COTTIN-DESGOUTTES REAR SUSPENSION BY CROSS SPRINGS.

brake drums are located on opposite sides of the center housing, hence the brakes are spring-suspended, which further reduces the amount of unsprung weight, and incidentally simplifies the brake linkage. The construction in general is rather complicated, however, requiring eight separate springs for the rear end, and four universal joints, each of the latter of a torque capacity about twice that of the joint ordinarily used in the propeller shaft.

Swinging Axle Tubes—A type of construction that has achieved a considerable degree of popularity in Central Europe provides axle tubes which, instead of being rigidly fastened into the center housing, are capable of swinging with relation to this housing, in a vertical plane, the housing itself being mounted on the chassis frame. The pivotal connection between the axle tubes and the center housing usually has sufficient span to give the necessary support to the tubes to enable them to withstand the moments due to driving and braking forces. Since all moving parts at the center of the axle are enclosed, it is necessary to fit the brake drums to the wheels. With this construction, even more than with the second type, there is a tendency for the wheel tread to change with spring play; under the deflection of the springs the wheels naturally tend to move apart at the bottom and draw together at the top. To reduce this variation in tread, some designers place the fulcrum of the axle tubes lower than the axis of the center housing or differential. Evidently this third type does not reduce the unsprung weight as much as the first two types described, but the general appearance is more conventional, and there are no exposed rotating parts underneath the car, as with the second type.

A good example of an axle with swinging tubes is the rear axle of the Tatra car made in Czechoslovakia. The center part of this axle is shown in horizontal and vertical section in Fig. 3. Another peculiar feature of this car is that, instead of the conventional pressed-steel frame, it has a tubular backbone extending from the engine back, and connecting to the driving-gear housing as shown in the illustration. The tubular backbone is provided with outriggers on which the body is supported.

This car has a spur-gear-type of differential, which is mounted on the propeller shaft instead of on the rear axle. This construction was used by the German Daimler Motor Company for many years in order to make possible the use of an "arched" rear axle, the two axle tubes both making a slight angle with the horizontal, so that dished wood wheels could be used and the spokes at the bottom, which carry the load, would still be vertical. Placing the differential gear

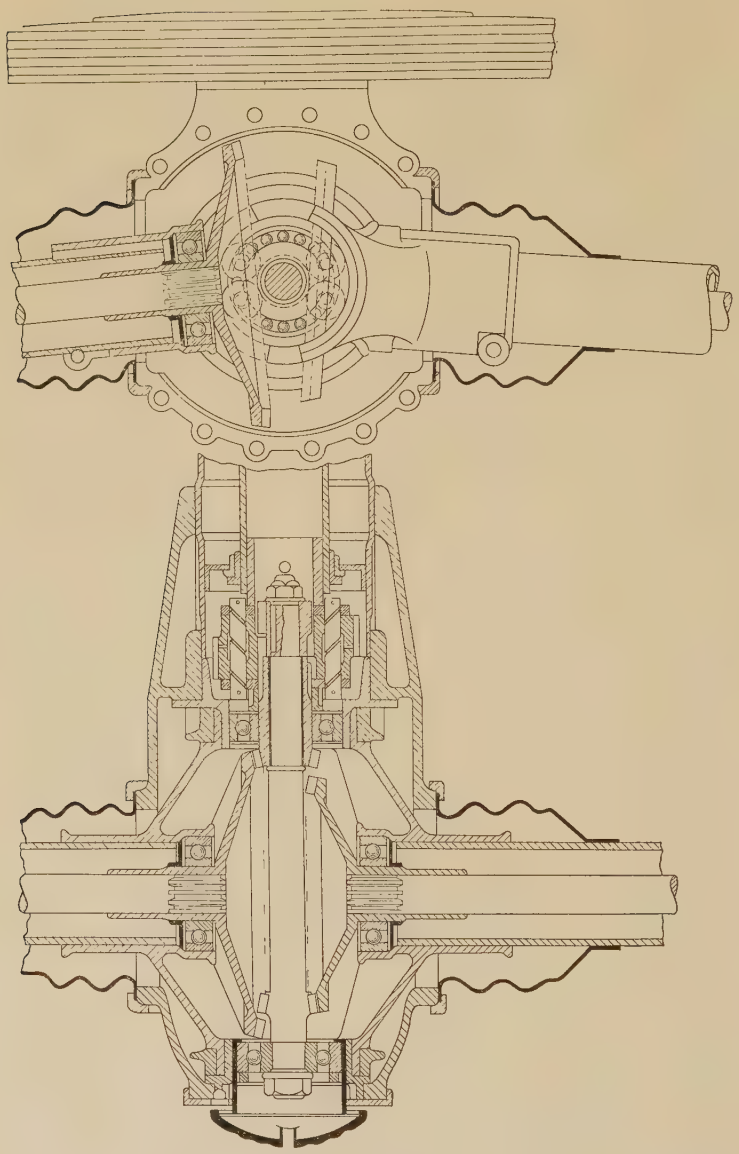


FIG. 3.—TATRA REAR CONSTRUCTION WITH SWINGING AXLE TUBES.

on the propeller shaft makes it necessary to use a separate pair of bevel gears for the drive to each rear wheel, and the two pairs must be of different dimensions, in order to avoid interference, but must have the same ratio.

Each axle tube at its inner end is provided with a two-armed spider or fork, the ends of the arms or prongs of which are in the form of cylindrical sectors which have a bearing between an inner cylinder fastened into the center housing and an outer ring slipped over them, this outer ring being provided with a radial flange to increase its rigidity. There are comparatively large openings in the center housing at both sides, through which the axle tubes extend, and these are closed by leather boots. The single spring is secured to

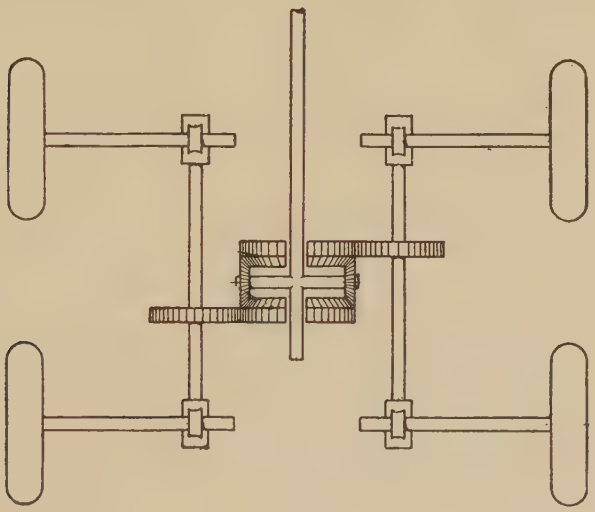


FIG. 4.—DIAGRAM OF DRIVE OF DUERKOPP SIX-WHEELED BUS.

the top of the center housing, and each of its free ends bears against a roller carried at the top of the brake backing plate.

The swinging-axle-tube idea has been worked out in a somewhat different form by Duerkopp in Germany, in a six-wheeled commercial vehicle chassis. In that case, too, the differential gear is mounted on the propeller shaft, and each side gear of the differential has a spur pinion fastened to it, which meshes with a spur gear on a shaft running parallel with the propeller shaft. There are two of these short shafts, and each carries two worms, which mesh with worm gears

above them. The axle tube is rigid with the worm-gear housing and swings around the axis of the worm. The general arrangement is shown diagrammatically in Fig. 4.

Cranked Axles—The fourth method consists in mounting a heavy crank on the chassis frame, on the projecting pin of which the wheel is mounted. This plan has been followed especially in Germany with commercial vehicles. One design, which is combined with a pneumatic suspension system, is illustrated in Fig. 5. A heavy crank has two bearings on

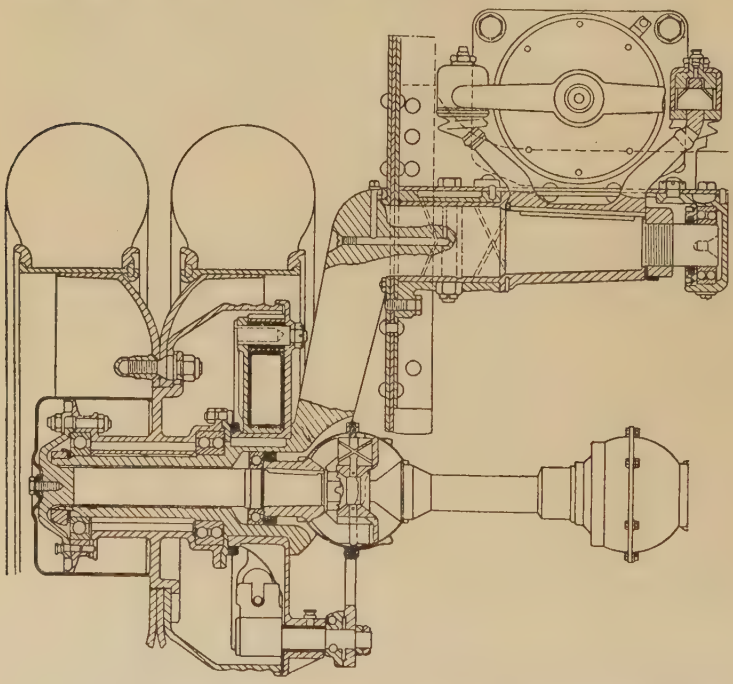


FIG. 5.—CRANKED REAR AXLE OF DAAG BUS CHASSIS WITH FAUDI PNEUMATIC SUSPENSION.

the chassis frame, the outer bearing being of the plain, and the inner one of the ball type. The crankpin forms the stub axle and is made hollow, so the driving shaft can pass through it. At its outer end this shaft is provided with a driving flange that is bolted to the wheel hub after the fashion of three-quarter-floating axles. Drive is from a frame-mounted center housing through a driving shaft with two large universal joints. The pivot portion of the crank is formed

with two arms which connect to a piston in a pneumatic cylinder which takes the place of the usual steel springs.

A similar method of independent springing is used at the rear end of the Vomag bus, which has front-end drive. In view of the fact that the crank arm is subjected to great torsion, owing to the overhang of the wheel spindle, it is made of tubular form, as shown in Fig. 6. The two bearings for the crank in this case are located on opposite sides of the crank arm, and they are very rigidly supported.

With six-wheeled vehicles the independent suspension problem can be worked out in a somewhat different way, as

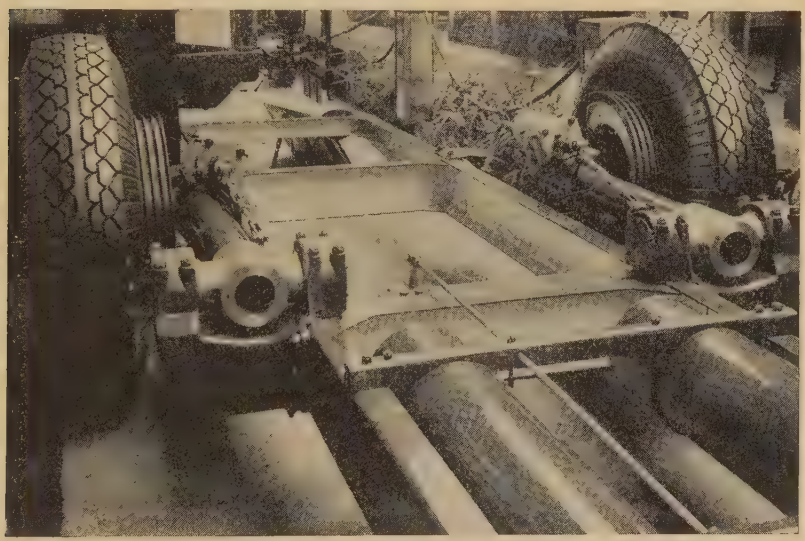


FIG. 6.—REAR CRANKED STUB AXLE OF VOMAG FRONT-DRIVEN BUS.

shown in Fig. 7, which illustrates the rear construction of the N. A. G. six-wheel bus. This chassis has a single dead axle at the rear, midway between the two pairs of wheels, which is carried by the frame. Upon each end of this axle is swiveled a casting, apparently of malleable iron, in the form of a double-armed lever with a hollow wheel-spindle at the end of each arm. Drive is by worm and wheel, and the two worm-gear housings are supported on the frame. From each housing extend two shafts with universal joints, each connecting to one of the drive wheels. It will be seen that with this construction each wheel can rise and fall independent of the others.

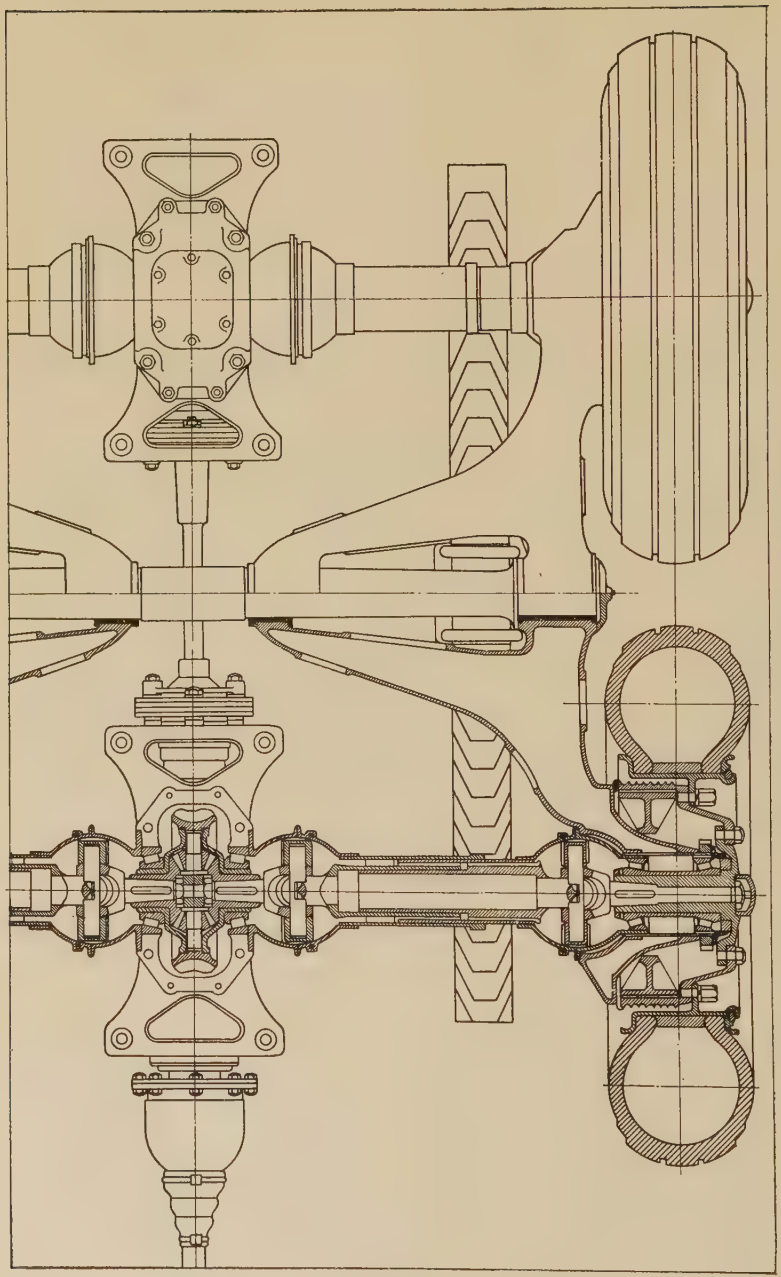


FIG. 7.—REAR CONSTRUCTION OF N. A. G. SIX-WHEELED BUS CHASSIS.

CHAPTER XVIII

Road Wheels

A considerable number of different types of wheel are used on motor vehicles. The wood wheel was taken over from the carriage and wagon industry, and the wire suspension wheel from the bicycle industry, but both of these types had to be modified considerably in design to suit them to the requirements of powerful motor-propelled vehicles, in which, in addition to carrying loads, they must also transmit the driving effort. Disc wheels, which were specially developed for motor-vehicle use, are made in two types, viz., rolled of malleable material, for use chiefly on passenger cars and buses, and cast, in which form they are used mainly on trucks. The ordinary cast wheel is made of steel or malleable iron, but abroad cast wheels of aluminum and magnesium alloys are used to a certain extent, and are, of course, adapted for use on light vehicles. Then there are spoked cast wheels, also made chiefly of steel and malleable iron, and used mainly for trucks; and finally there is the pressed-steel spoked wheel, which is made in several forms. In England a wheel of this type is largely used for passenger cars which is very similar to a wood wheel in appearance, the entire wheel, including hub, spokes and felloe, being made in two pressings, which are united by seam-welding.

Wheel Diameters—The outside diameters of peumatic-tired wheels for passenger cars vary from 28 to 33 in., substantially as the weights of the cars. The rim diameter, which usually is less than the wheel diameter by twice the nominal width of the tire, varies from 18 to 21 in.

Wheel diameters for buses with pneumatic tires average as follows:

12–16 passengers.....30 in.	23–30 passengers.....34 in.
17–22 passengers.....32 in.	31–42 passengers.....36 in.
Over 42 passengers.....38 in.	

Motor trucks equipped with pneumatic tires have wheels of the following diameters on the average: One ton, 30 in.; 2 ton, 32 in.; 3 ton, 34 in.; 4 ton, 36 in. Wheels of solid-tired trucks are made 34 in. in diameter for capacities of 2 tons and less; 36 in. for capacities of 2½ to 5 tons, and 40 in. for capacities of over 5 tons. The diameters here given are averages and there are, of course, variations from them in individual cases. Front and rear wheels are always made of the same diameter, so that the same spare tire will serve for both. In buses and trucks, where the maximum load on the rear wheels is usually much greater than that on the front wheels, it is customary to use dual tires in the rear and singles in front, all of the same size.

Tire diameters have decreased materially since the earlier days of automobile development. With the smaller wheels, the stress on the axles due to lateral impacts are less, as they are directly proportional to the wheel diameter. Perhaps the chief reason for the use of smaller wheels is that they make it possible to lower the center of gravity, which is desirable for reasons of safety. The lower cost of the smaller tires and the fact that with high-speed engines a large reduction ratio would be required if large wheels were used, which it is difficult to obtain in a single step, probably also had some influence on the trend. What limits the minimum wheel diameter is the need for a certain minimum road clearance on rutted roads.

Demountable Wheels and Rims—In order that tire punctures or blowouts on the road may not cause any serious delays, pneumatic-tired vehicles are now equipped with one or more spare tires, fully inflated, on either demountable rims or demountable wheels. In case of a tire defect, the spare rim or spare wheel is substituted for the rim or wheel carrying the defective tire. The demountable rim was developed chiefly in the United States and is in most extensive use here, while the demountable wheel has met its greatest popularity abroad.

The advantages of the demountable rim are that it weighs less than a demountable wheel, hence the spare represents less of a load, and can be more easily handled in making tire replacements; and that, the means of fixation, being further out from the axis, are subjected to less strain and therefore need not be so strong. An advantage of the demountable wheel is that its rim is lighter, and it therefore has less flywheel effect, which counts especially in racing, as it permits of quicker acceleration and deceleration. It is natural that the lightest type of wheel, the wire suspension wheel, should be used mainly in the form of a demountable

wheel, but the demountable feature is used also in connection with rolled disk wheels and wood wheels.

Wood Wheels—The type of wood wheel used for motor vehicles is known as the artillery wheel, which is characterized by the fact that the mitered heads of the spokes are clamped between a radial flange formed on the metal hub and a corresponding flange slipped over the hub. In most cases the pressed steel brake drum performs the function of this loose flange. Formerly not only the spokes but also the felloes were made of wood, but wood felloes, which require a particularly high quality of wood, which must be steamed and bent in forms in the production process, have been almost entirely replaced by steel felloes. The steel felloe is made of channel section, one of its flanges having an extension which is turned outward at an angle to act as a support for the tire rim. Holes are punched in the bottom or web of the channel to receive tenons on the ends of the spokes. These tenons are made with a large radius at their junction with the spoke proper, so as to provide a substantial shoulder for the felloe and at the same time offer great resistance to shearing stresses. Tenon diameters are made in accordance with the following table, which is an S. A. E. Recommended Practice:

	Inches	Inches	Inches	Inches
Tire width	3½ or 4	4	4½	5
Spoke size	1¼	1⅜	1½	1¾
Tenon-hole diameter	½	⅝	⅝	¾

The width of the spokes may be either constant or taper down slightly from the hub to the felloe. In truck wheels for solid tires (which generally have felloes that are quite wide), in order to give adequate support to the felloe the thickness of the spoke is sometimes made to increase toward the felloe. The spokes for truck wheels, moreover, are often made of substantially rectangular, instead of round or oval section.

The length of the miter or head of the spoke should be about 1.40 times the thickness thereof. The throat of the spoke, or that portion intermediate between the barrel and the miter, is drawn to a radius of about 2 in. and so that the throat circles of adjacent spokes intersect ⅜ in. from the hub flange circle, while the curved edge of the face of the miter comes 3/16 in. from the hub flange circle.

Figure 1 illustrates a wood-spoked front wheel for a car weighing about 3000 lbs. This wheel has twelve spokes with a 1 9/32 by 1 7/16 in. section at the inner end and a 1 7/32 by 1 5/16

in. section at the outer end of the barrel, the first dimension in each case being that in the direction of the wheel axis. The axial width of the miter is 1⅜ in. In this wheel there are as many hub bolts as there are spokes. In many twelve-spoked wheels only nine or six hub bolts are used.

The bolt holes in the hub flange are either broached out to a square section and the bolts given a corresponding section directly below the head, or else the bolts are formed with serrations below the head and forced through holes of slightly smaller diameter, so that the serrations grip the metal of the

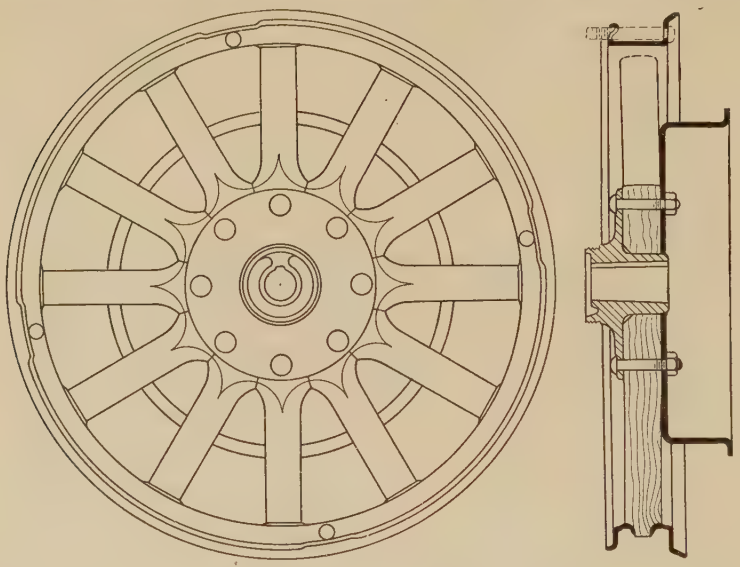


Fig. 1.—Passenger-Car Wood Wheel.

flange, thus preventing the bolt from turning when the nuts are screwed home. In most designs of wood wheel the brake drum serves as the loose flange, and sometimes the inner portion of its web is offset, so the outer portion will be at some distance from the spokes, which prevents the accumulation of dust and mud in the sharp corners between the spokes and the brake drums.

Truck-Type Wood Wheels—Figure 2 shows a sectional view and a side view of a truck wheel with wood spokes and wood felloe, the spokes being of rectangular section. A steel felloe band is shrunk over the wood felloe. Such wheels, of

course, are generally fitted with solid rubber tires. These are vulcanized onto a channel-section base band with a corrugated surface, and the tire with its base band is forced over the felloe band under great pressure in an hydraulic press, which assures such a tight fit that the friction between the two bands prevents any displacement of the tire under the forces acting on it in service. In the case of cast wheels the solid tire with its base band is fitted in a similar manner directly to the machined rim of the wheel. To ensure a proper fit, the dimensions of both bands must be held within close limits, and these have been standardized by the S. A. E.

Wood-spoke, metal-felloe wheels of a special design are made for trucks, in which steel caps are forced over the ends of the spokes, these caps having tenons on them which are riveted into the steel felloe. This construction results in a stronger wheel, as the compressive stress on the spoke is distributed over its entire section, and the entire section of the spoke is available also to resist shearing stresses.

Advantages and Disadvantages of Wood Wheels—Among advantages claimed for wood wheels are their ability to absorb shocks due to the elasticity of the wood, freedom from flaws and hidden weak spots, ease of repair, lightness in proportion to strength, and low cost of manufacture. A disadvantage of wood wheels is their sensitiveness to atmospheric influences. In very dry climates they are apt to shrink and come loose in the hubs.

Wood wheels, as a rule, are shipped by the wheelmaker to the car or truck manufacturer without the hub. Before leaving the wheel factory the wheel has received a protective priming coat, and wheelmakers recommend that another coat be applied immediately upon receipt at the truck factory, which will protect the wheel while in storage and during road tests. Without the hub the wheel assembly, of course, is comparatively weak, and great care must be taken not to injure it during transit. Wheels should always be stored in dry, cool rooms with a minimum temperature variation. Hubs should be fitted by hand and should be neither a loose fit nor unduly tight. Great force applied in assembling the wheel on the hub is almost certain to distort the wheel spider. All nuts on hub bolts must be set up tight, and light tapping of the head of the bolt after the nut has been screwed home is an aid in making sure of a proper set-up. After the tapping, the wrench should be applied to the nut once more, and then the threaded end of the bolt riveted over.

Fastening Brake Drum to Spokes—As a rule the only means for fastening the brake drums to the wheel are the

hub bolts, but in the case of wheels and drums of considerable diameter it is sometimes considered desirable to connect the spokes to the brake drum at a greater distance from the axis

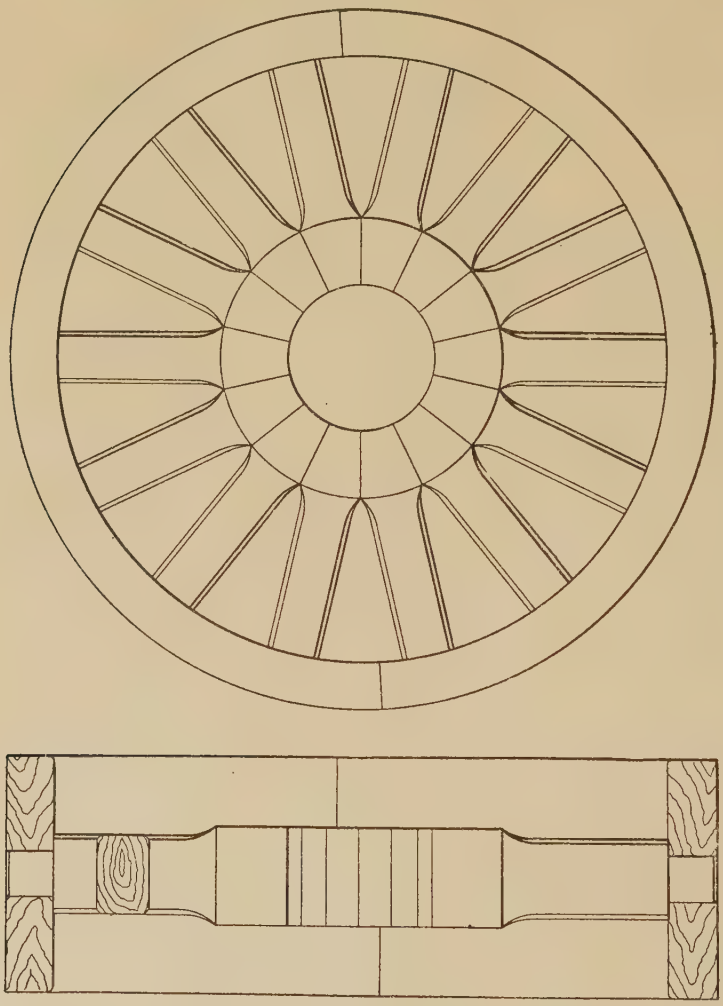


Fig. 2.—Spoke and Felloe Assembly of Truck Wheel.

of the wheel, so as to reduce the strain which must be withstood by the fastening means. There are two possible methods, and both are illustrated in Fig. 3. At A the spoke is shown to be formed with a boss which is drilled for a bolt that

connects the spoke to the brake drum. Usually only every other spoke is thus bolted to the brake drum, the intermediate spokes being without the boss. At B the spoke is shown fastened to the brake drum by means of a clip.

Wire Wheels—The wire suspension wheel is the lightest wheel made, and that it is not used more extensively is due to its greater cost and to the greater difficulty in cleaning it. It is claimed to be more flexible or yieldable than other types of wheel, and it appears to be generally admitted that it is less severe on tires, which no doubt is due mainly to its lesser weight, but may be due also in part to this greater flexibility.

In the original wire wheels, as used for bicycles and as used today for motorcycles, the spokes are threaded through holes in two flanges formed on the hub, the button-head end of the spoke being turned almost at right angles. In automobile wheels, however, the spokes are threaded through holes

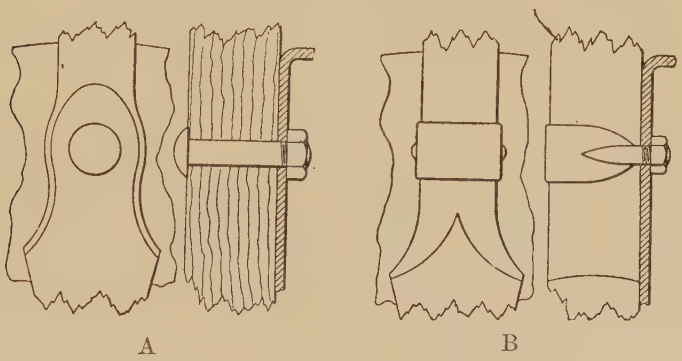


Fig. 3.—Two Methods of Fastening Brake Drums to Wood Spokes.

drilled in the hub shell itself, and the ends are bent no more than required by the tangential "lacing."

There are four methods of lacing wire wheels, as illustrated in Fig. 4. At A is shown the ordinary two-row lacing as used in motorcycle wheels, the two rows connecting to the hub at opposite ends and to the rim at some distance on opposite sides of the center plane. It may here be explained that the feature which distinguishes the wire wheel from other types is that the spoke material works entirely in tension. The spokes, of course, must take up not only the radial stresses due to the static load on them, but lateral stresses on the wheel due to shocks from both sides, and tangential stresses

due to driving and braking forces; and the lacing must be such that for any of these stresses there are spokes which are subjected to tension. Consequently, with a two-row lacing all of the spokes, as a rule, are arranged tangentially, one-half of each row in one direction and the other half in the opposite direction. In order that the spokes may afford due strength against lateral stresses, each row must make an appreciable angle with the center plane of the wheel. The stresses from the outer side are always the greatest, and the outer row of spokes, which takes these stresses, is inclined the most.

At B is shown the crossed lacing which seems to make

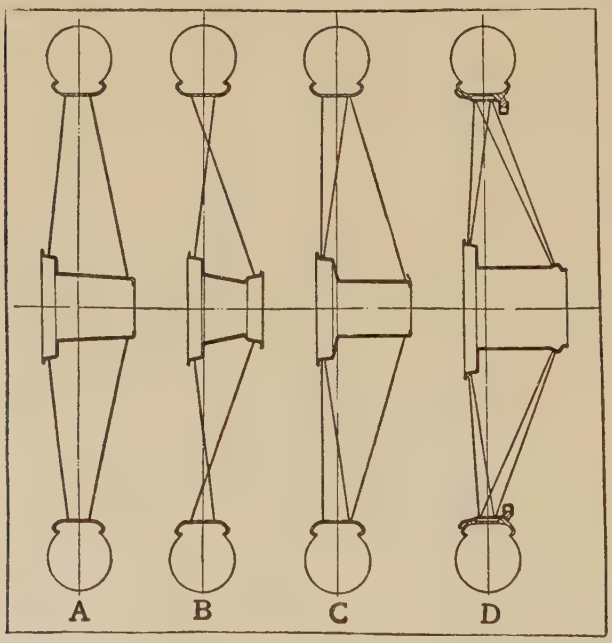


Fig. 4.—Methods of Lacing Wire Wheels.

possible the use of a shorter hub. C shows the triple lacing, which is the one in most extensive use, and D shows a quadruple lacing. In triple-laced wheels one set of spokes extends from the outer end of the hub to the outer side of the rim; a second set, from the inner end of the hub to the outer side of the rim, and a third from the inner end of the hub to the inner side of the rim. The two sets connecting to the inner end of the hub take all driving and braking stresses, while

the set connecting to the outer end of the hub takes stresses due to skidding.

Wire Spokes—The spokes of laced wire wheels are made of No. 1045 or a similar steel, the strength of which is considerably increased by swaging. The Wire Wheel Corp. of America uses a spoke made of wire of 0.205 in. diameter. This is reduced by swaging so it tapers from the original diameter at the head end down to 0.148 in. near the middle. The threaded end is then increased in size to the original diameter,

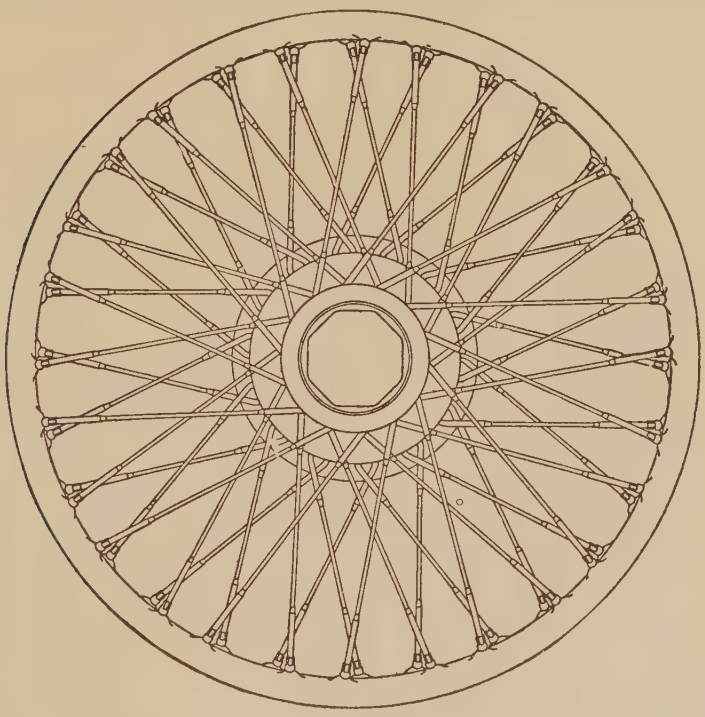


Fig. 4-a.—Side View of Wire Wheel with Tangential Lacing.

the diameter also decreasing very gradually from this end toward the middle. Having the spokes thinnest at the middle and varying the section very gradually was found to prevent concentration of stress due to vibration of the spokes, and consequent breakage. The head is made in "bung" form, tapering down from a maximum diameter of 0.315 in. at an angle of 25°. Seats for the heads in the hub shell are reamed

to give an exact fit. To the rims the spokes are secured by means of threaded nipples made of soft steel, and holes are punched though the rims at the bottoms of spherical depressions, so shaped that the heads on the nipples will have a "square" seat.

Driving Means—The triple lacing was originated in England by John V. Pugh of the Rudge-Whitworth Company, and wire wheels became popular first in that country. These wheels always have been made as demountable wheels and are provided with a hub shell which passes over an inner hub

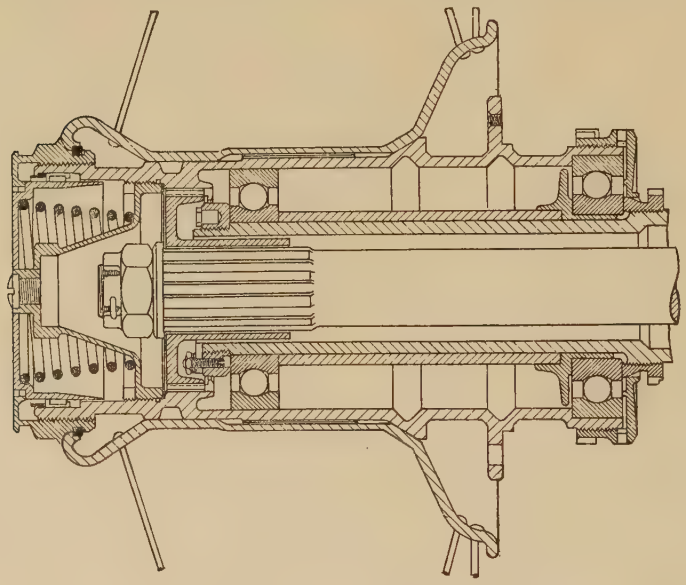


FIG. 5.—HUB OF RUDGE-WHITWORTH WHEEL FOR ROLLS-ROYCE CAR.

secured to the axle shaft. Means must therefore be provided not only for holding the wheel concentric with the inner hub, so it may run true, but also for driving it. Figure 5 shows one way of accomplishing this, the hub shown being that of the Rolls-Royce car. At the inner end the hub shell rests on a tapered seat on the inner hub, against which it is forced by a hub cap screwing over the inner hub at the outer end. The drive is taken by serrations at the middle on the outer surface of the inner hub and the inner surface of the hub shell. Beyond the serrations the hub shell is guided on a cylindrical portion of the inner hub, and at the outer end it

is provided with a turned-edge against which a suitably formed surface of the hub cap bears.

Positive Locking Means—With all demountable wheels it is essential to have a positive locking means, so that the wheel cannot possibly come off when the car is in motion. The wheel shown in Fig. 5 has such a lock. It will be noticed in the drawing that within the outer end of the inner hub there is a sliding barrel which has splines or keys on its outer side, engaging with splines on the inside of the inner hub, so it cannot turn relative thereto. The outer end of this sliding barrel projects through the hub cap and carries keys which engage into key slots in the hole in the hub cap. The

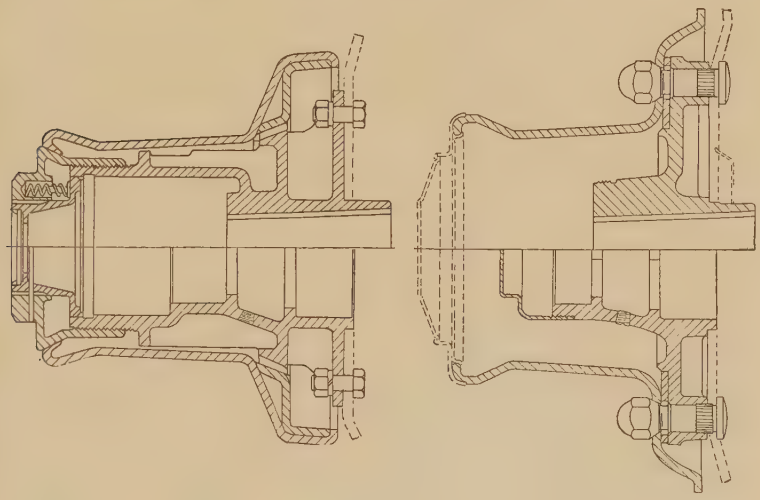


Fig. 6 (left).—Buffalo Corrugated Taper Drive.
Fig. 7 (right).—Hub of Bolted-on-Type of Wire Wheel.

barrel is forced outward by a heavy coiled spring which seats against a ring nut screwed into the inner hub. To unscrew the hub cap it is necessary to force the sliding barrel inward, which is accomplished by means of a special wrench applied to the hub cap. The square-section ring in a groove in the hub cap serves to pull the wheel off with.

A somewhat modified design is used by the Wire Wheel Corp. of America. It is known as the corrugated taper drive, and is illustrated in Fig. 6. The inner hub is formed with a driving flange in which there are 18 corrugations on a 20°

taper, which fit into corresponding corrugations in the hub shell. Corresponding to the taper on the hub flange is a taper on the hub cap, which, when screwed into place, draws the hub up on the driving flange and seats it on the corrugations, thus wedging the wheel between the two tapered surfaces.

Another driving method is represented by the Buffalo bolted-on-type of wire wheel, the hub of which is shown in Fig. 7. There are six holes in the hub shell, corresponding to six drive studs fastened to the inner hub. The wheel is held in place by six acorn nuts. This type is cheaper to manufacture than the corrugated-drive type, and is also somewhat lighter. These wheels are also made with the driving studs concealed in the hub shell.

Buffalo wire wheels are secured by a positive lock cap similar in principle to the one already described. The sliding barrel in this case is held under tension by four coiled springs, and is operated by a special wrench having a puller and cam lever. To lock the wheel on the hub by the hub cap, the sliding barrel is withdrawn by the special wrench. The hub cap is then screwed in place and the sliding barrel released, thus permitting the teeth on the inner end of the sliding barrel to snap into engagement with teeth on the hub itself.

Ford Wire Wheels—A special design of welded wire wheel is used on Ford cars. The wheels have two rows of spokes which are crossed and are welded to both the hub shell and the rim, so the entire wheel is in one piece. There are thirty spokes, of ¼ in. diameter, and of a tensile strength of 4000 lbs. each. The spokes in the outer row are radial and designed to take radial and lateral loads only, while those in the inner row are crossed and take tangential (driving and braking) loads in addition. The hub shell fits over an inner hub, and, together with the brake drum, is bolted to a flange on this hub. Its inner end is of nearly the same diameter as the brake drum, which makes the spokes of the inner row comparatively short. The rims of these wheels are of the so-called drop-center type, which permit of the use of tires with inextensible beads on non-collapsible rims without removable flange.

Disc Wheels—The scarcity of wood suitable for wheels led the Michelin Tire Company of France about 1915 to produce a wheel in which the hub and felloe were connected by a single dished disc of steel. In order to combine a high degree of lateral strength with lightness, it was necessary to decrease the thickness of the disc from the hub flange outward, and in the early Michelin disc wheels the discs were

turned in the lathe. In this country the Michelin wheel was taken up by the Budd Wheel Company, and machinery was developed for tapering the discs by a rolling process.

In the Budd-Michelin disc wheel the disc is secured to the hub flange by means of four, six or eight studs, the latter number being used only in wheels for heavy commercial vehicles. The studs are permanently secured into the hub flange. It was found undesirable to have the disc ride directly on the studs, hence it is pressed with a ridge around the

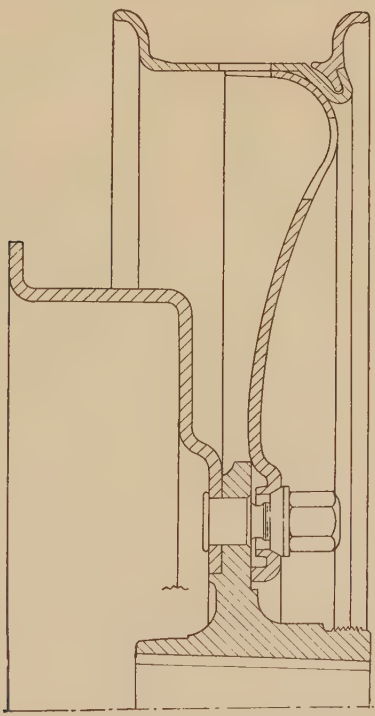


Fig. 8.—Sectional View of Budd-Michelin Disc Wheel.

stud-holes, as shown in the sectional view Fig. 8. Ball-faced bronze cap nuts engage the "coined" ridges around the stud holes and center the disc. Thus all forces acting between the disc and the studs are transmitted through the bronze nuts, and the flexibility of the steel in the coined ridges has the effect of locking the nuts so they will not come loose.

Right- and left-hand threads are used for the studs and nuts on the respective sides of the car, which is said to be a further safeguard against coming loose of the nuts. There is considerable clearance around the studs in the holes in the disc, which facilitates removal and replacement of the wheels. At

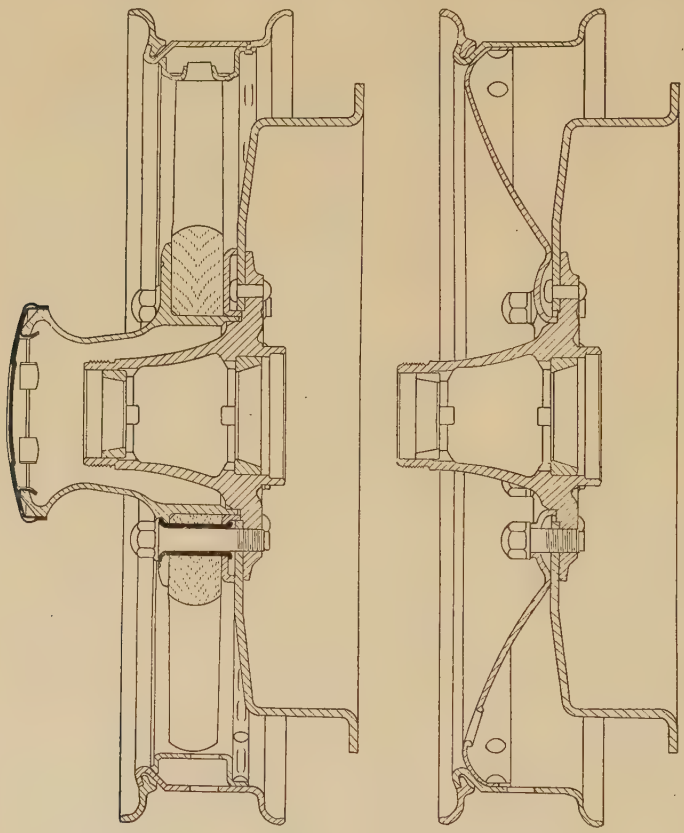


Fig. 9 (left).—Demountable Wood Wheel.
Fig. 10 (right).—Disc Wheel Interchangeable with Wood Wheel.

the rim, the disc is flanged and secured by riveting. In the original French wheel the rims were of the clincher type, but wheels made in this country have rims with a removable flange, which take straight-side tires.

Interchangeable Disc, Wood and Wire Wheels—Many car manufacturers equip certain of their sports or de luxe models with wire wheels, while the standard models carry wood or disc wheels; or wire wheels are offered as an option while wood wheels are standard equipment. The problem then arises of designing the various wheels so that one type can be replaced by another without too much trouble. To meet this situation, the Motor Wheel Corporation designed a set of four demountable wheels all interchangeable on the

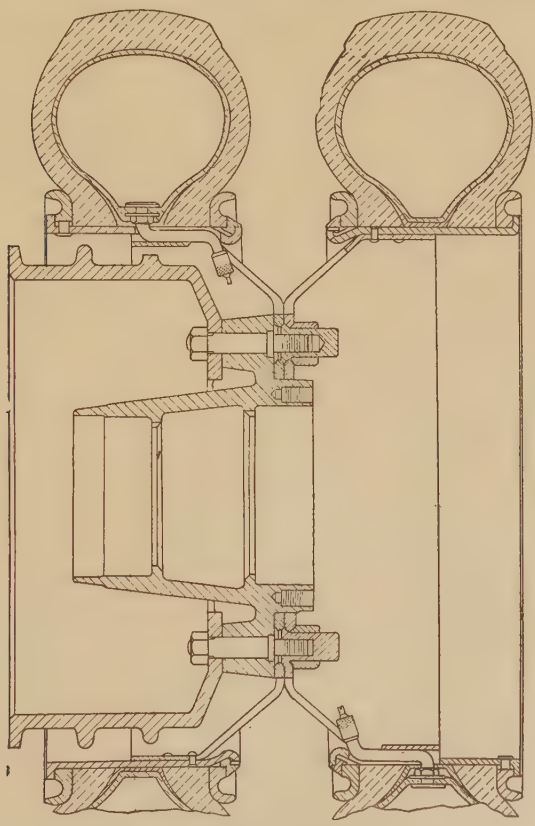


FIG. 11.—DOUBLE-DISC-TYPE OF WHEEL.

same hub—the demountable wood wheel shown in Fig. 9, the disc wheel illustrated in Fig. 10, and two types of wire wheel, the difference between the two designs being that one has a

hub shell of much larger diameter than the other and has the heads of the driving cap screws concealed inside the hub shell, while in the other they are exposed. These wire wheels have

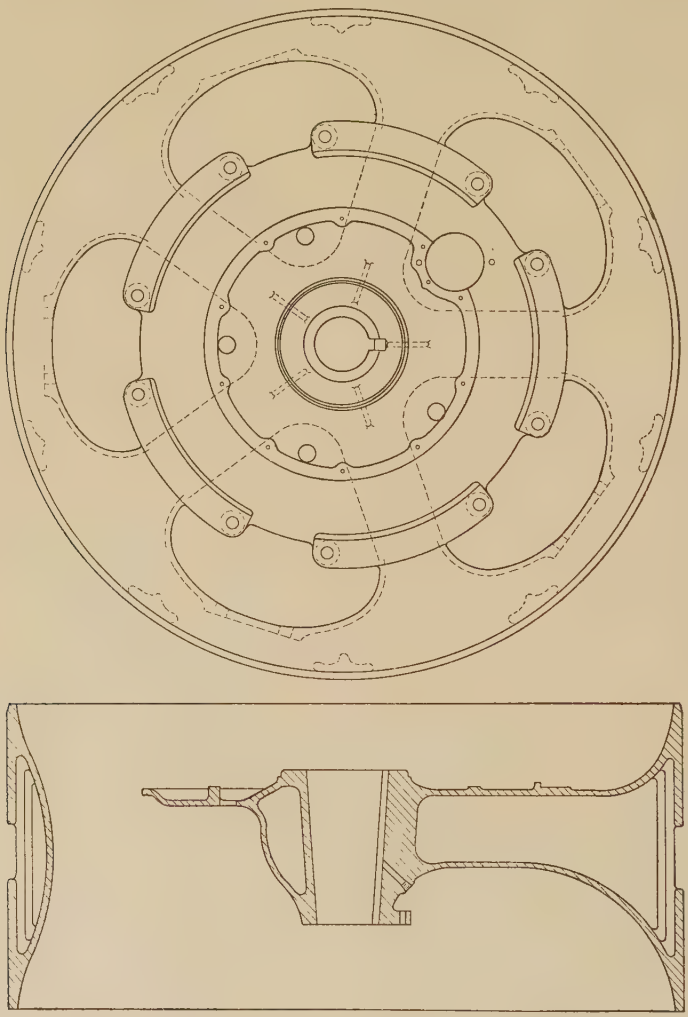


Fig. 12.—Cast-Steel Truck Wheel for Solid Tire Equipment.

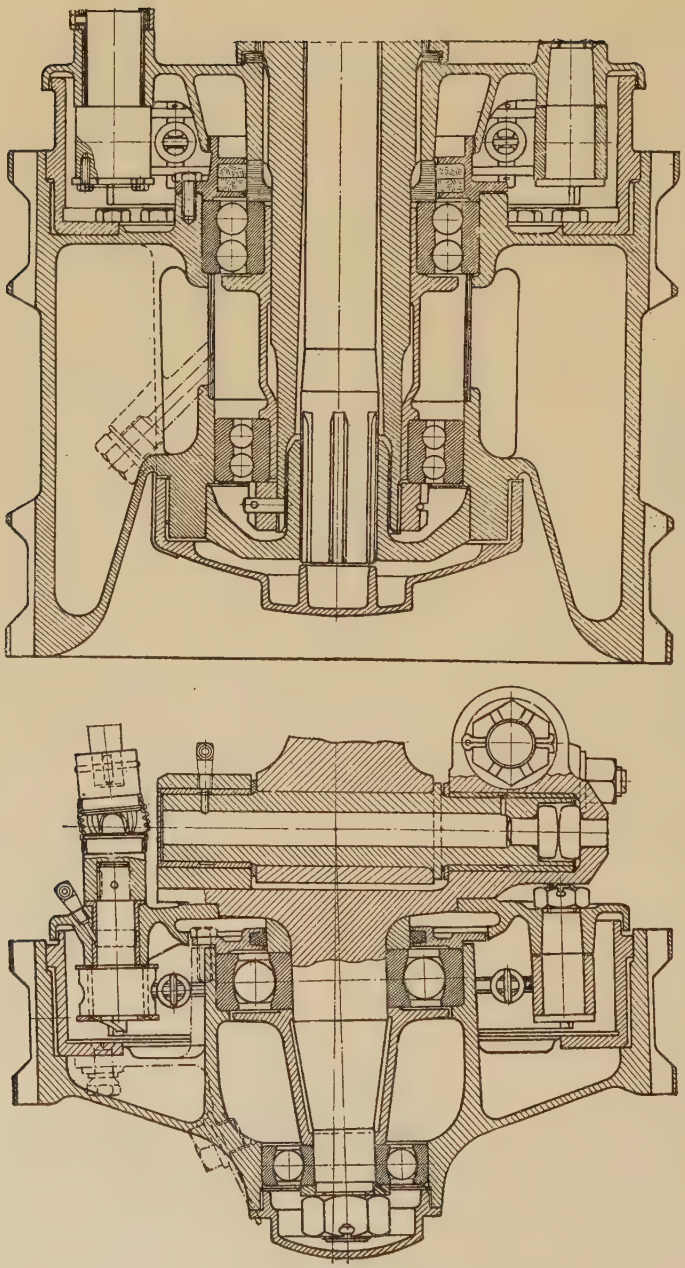
a crossed lacing. Different retaining bolts are needed, but otherwise the wheels are completely interchangeable on the same hub.

Dual Disc Wheels—Disc wheels are much used on motor buses, which usually carry single wheels (and tires) in front and dual wheels in the rear. Both sets of wheels are deeply dished. The disc of the front wheel is secured to a flange near the outer end of the hub, to make room for a brake drum within the wheel. At the rear the two discs are secured to the same hub flange, near the middle of the hub. A rear wheel of this type is illustrated in Fig. 11. In all disc wheels in which the disc fastens to the outer side or to the center of the rim, it is necessary to cut an opening in it for the tire valve. In the dual wheel shown in Fig. 11 this applies to the inner wheel, and an opening is needed in the outer disc also to get access to the tire valve.

Cast Wheels—Wheels cast of steel or malleable iron are largely used for trucks and buses. There are two general types of cast wheels, the disc and the spoked type. The spoked type may be again divided into two sub-classes according to whether the spokes are solid or hollow. Where solid spokes are used they generally have a cruciform or Y-section, but the hollow-spoke type of wheel is most popular. Naturally, a substantially round, hollow spoke has greater strength than a solid spoke of equal cross-section, and it also has an advantage from the standpoint of appearance. With a wheel cast in a single piece from either steel or iron, it is not difficult to get sufficient strength, but it is quite difficult in wheels for the lighter vehicles to keep the weight within permissible limits. This is due to the fact that, in order to ensure sound castings, the thickness of the metal cannot be reduced below a certain minimum. The best distribution of metal in such a wheel is quite a problem. In one recent design the rim is made of a rather deep box section, so it is not dependent upon the spokes to any extent for radial stiffness, and only three spokes are used. More commonly, however, the spokes number from five to seven, while some designs for heavy trucks have as many as ten spokes.

One advantage that is claimed for cast metal wheels is that, barring severe collisions, they do not lose their shape in service, and that since they stay round permanently they are easier on tires.

Cast metal wheels were not used in this country to any extent previous to the war. In connection with the design of the Class B Army truck some tests on wood and metal motor truck wheels were made at the Bureau of Standards, and the results, with data on the wheels, are given in Technologic Paper No. 150 of the Bureau. The average com-



FIGS. 13 and 14.—EUROPEAN CAST STEEL FRONT AND REAR WHEELS FOR PNEUMATIC TIRES.

position of the steel in the cast-steel wheels tested by the Bureau was as follows:

	Per cent		Per cent
Carbon	0.25	Phosphorus	0.042
Silicon	0.38	Sulphur	0.060
Manganese	0.68		

The Dayton Steel Foundry Company, a large manufacturer of cast-steel wheels, uses electric furnace steel of the following specification:

	Per cent		Per cent
Carbon	0.20–0.25	Phosphorus	0.040 max.
Manganese	0.60–0.75	Sulphur	0.045 max.
Silicon	0.23–0.28	Nickel	0.05–0.20

The above steel has the following physical properties:

Ultimate strength	68,000 lbs. per sq. in.
Yield point	42,000 lbs. per sq. in.
Elongation	23 per cent
Reduction of area	28 per cent

Figure 12 shows a rear wheel for a 2-ton truck, manufactured by the Dayton Steel Foundry Company. The wheel has five hollow-arch-section spokes and is designed to receive a 36 by 14-in. solid tire. Both the felloe and the hub are of box section, and the metal in the spokes is ¼ in. thick, while the inner wall of the hub and the edges of the rim are ⅜ in. thick. The weight of this wheel is 305 lbs.

In this connection it may be of interest to state that the average weight of six metal wheels for 36 by 10-in. solid tires tested by the Bureau of Standards was 280 lbs., the average weight of two wheels for 40 by 10-in. wheels, 307 lbs. and the average weight of seven 36 by 5 in. wheels, 134 lbs. From this it would seem that with wheels for solid tires, for the same nominal diameter the weight of the wheel varies substantially as the tire width, while the carrying capacity of the tires increases considerably faster.

In the Dayton wheel the spokes join the rim with large radii, which gives a sort of arch effect. In order to get a good support for the cores in casting, the rim is cast open at the middle, but its two sides are connected by ten tie-bars. A disc with five machined sectors is cast integral with the wheel, by which the brake drum is carried. The outside of the rim is turned off smooth to size, and one edge is beveled off to receive the solid-tire base-band.

Cast-Steel Wheels for Pneumatic Tires—The tendency is constantly more toward the use of pneumatic tires, even on the heavier vehicles, and manufacturers of wheels for such vehicles have developed wheels adapted to these tires. Figures 13 and 14 show cast-steel wheels produced by the Fischer Steel Foundry of Switzerland, as fitted to the Fross-Buessing bus made in Vienna, Austria. These are 20-in. wheels and designed to take either 34 by 7-in. or 36 by 8-in. tires, a single tire for the front wheel and duals for the rear one. The design is self-explanatory, but attention may be called to the method of ventilation at the rim, to prevent excessive heating of the tires. The brake drums have good metallic contact with the wheels, so the heat generated at the drums will flow into the wheels and be dispersed thereby, while the tire rims are separated from the continuous wheel rim, and ventilating channels are provided between the two.

Malleable Iron Wheels—Many truck and bus wheels are made of malleable iron, that is, cast iron which has been partly decarbonized by a prolonged exposure to high temperatures. An average composition of malleable iron is as follows:

	Per cent		Per cent
Carbon	2.45	Phosphorus	0.200
Manganese	0.22	Sulphur	0.065
Silicon	0.80		

Figure 15 illustrates the Smith wheel, which is of malleable iron. The particular wheel shown is designed to take either a 30 by 5 in. or a 32 by 6 in. pneumatic tire, as used on trucks of 1¼ to 1½ tons capacity. It has five spokes and is of the hubless type, being secured to a separate hub by means of bolts. The hub portion is comparatively deep radially and is reinforced by five radial ribs midway between spokes, ⅛ in. thick. The walls of the substantially round spokes are 9/64 in. thick. The rim is of open channel form, but the flanges of the channel are braced by twenty-two ¼-in. round braces near the circumference. At one point of the felloe ribbed guides or stops for the driving flange on the rim are cast. When designed for solid rubber tires, the rim of the Smith wheel is made of box section, the inner and outer walls being tied together by numerous pillars of cruciform section.

The hubless construction, which necessitates a separate hub, has the advantage that the machining of the bearing seats, which must be very accurate, can be done to greater advantage. The weight of the wheel, of course, is slightly increased by the separate hub.

Magnesium Alloy Wheels—Wheels cast of a magnesium alloy known as Elektron are made in Germany by the I. G. Farbenindustrie A. G. in Frankfort-on-Main and are used by the Berlin General Omnibus Company as well as on many private passenger cars. Those used on the Berlin buses are a sort of modified disc wheel with radial corrugations. There

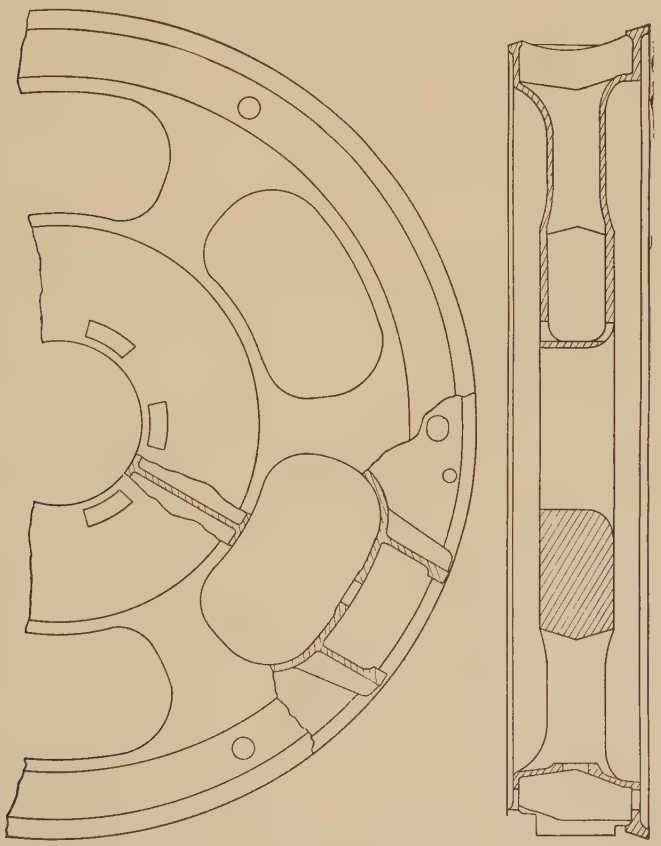


FIG. 15.—MALLEABLE IRON TRUCK WHEEL FOR SINGLE PNEUMATIC TIRES.

are six such corrugations in the front wheel, which weighs 48 lbs., and eight corrugations in the rear wheel, which weighs 92 lbs. Use of these wheels resulted in a saving of 65 per cent in weight, the actual saving for the whole bus amounting to 500 lbs.

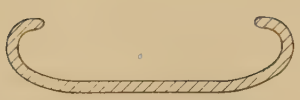


FIG. 16.—CLINCHER RIM SECTION.

FIG. 17.—STRAIGHT-SIDE RIM SECTION.

Types of Pneumatic Tire—There are two general types of pneumatic tire, the clincher and the straight-side type. The rims for the two types of tire are shown in Figs. 16 and 17. Clincher tires are manufactured with an extensible bead, so that the bead portion can be stretched when forcing it over the rim in removing and replacing the tire. When the tire is in place, the air pressure in the inner tube forces the bead portion of the cover into the clincher portion of the rim, thus preventing the tire from coming off the rim.

Straight side tires, which are made with non-extensible beads, require rims with a removable side ring or rims which are split at one point of their circumference and therefore are collapsible. The side ring also is held in place by the air pressure within the tire. Either the side ring itself is split and engages into a groove in the rim which makes a small angle with the plane thereof, or else it is continuous and slid over the rim, and is supported by a locking ring behind it, which is split and seats in a groove in the rim.

Straight-side tires made with non-extensible beads may, however, be used also with solid rims of a special design

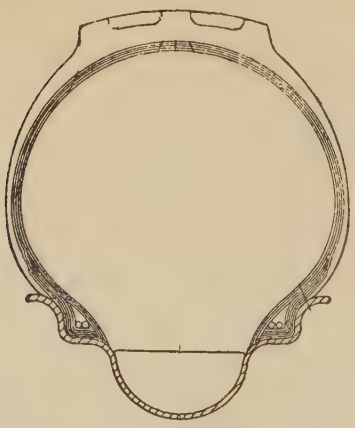


FIG. 18.—DROP-CENTER RIM.

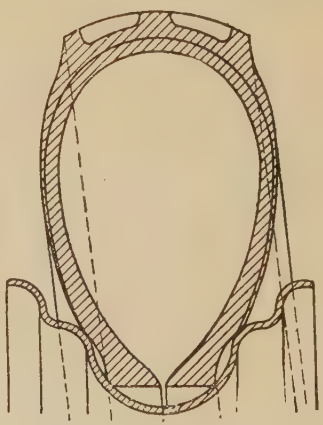
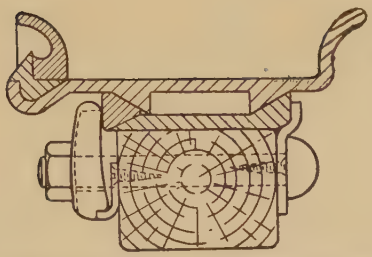


FIG. 19.—PLACING TIRE ON DROP-CENTER RIM.



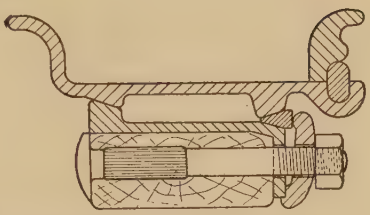
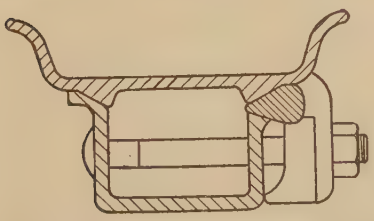


FIG. 20 (LEFT).—FIRESTONE DETACHABLE, DEMOUNTABLE RIM ON WOOD FELLOE.

FIG. 21. (RIGHT).—CLEVELAND DETACHABLE, DEMOUNTABLE RIM ON WOOD FELLOE.

known as the drop center type. As illustrated in Fig. 18, the central portion of the base of the rim forms a channel, and when it is desired to mount a tire on such a rim, the beads of the tire are placed in the drop center or channel at one side of the wheel (Fig. 19), which makes it possible to force the bead on the rest of the tire over the flange of the rim. When inflated, the beads of the rim expand against the flanges all around and, being inextensible, they cannot get off the rim. These drop-center rims were used for bicycle tires at one time, and are now coming into use on automobiles in connection with balloon or low-pressure tires.

Tire Rims—Two detachable, demountable rims for pneumatic tires are shown in Figs. 20 and 21, the first being a Firestone and the second a Cleveland. Both rims are rolled from special section steel with wedge-shaped projections on the under side. Each rim has a removable side ring and a locking ring, the latter being split. The locking ring is snapped into position between the rim and the side ring while



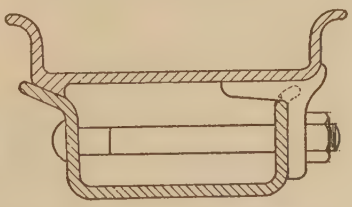
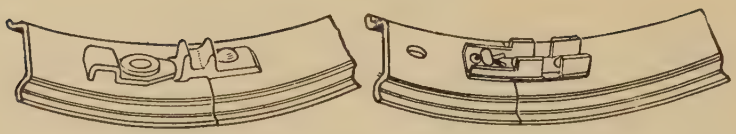


FIG. 22 (LEFT).—FIRESTONE COLLAPSIBLE DEMOUNTABLE RIM ON STEEL FELLOE.

FIG. 23 (RIGHT).—HAYES COLLAPSIBLE DEMOUNTABLE RIM ON STEEL FELLOE.

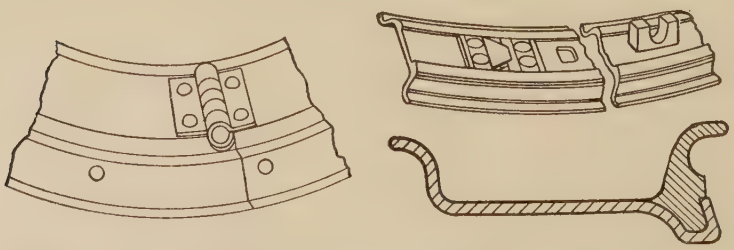


FIGS. 24 and 25.—SPLIT RIM LOCKING AND DRIVING MEANS (Jaxon).

the tire is deflated. Rims of this type are used chiefly for light commercial cars. The wedge-shaped rings on the base of the rim are seated on corresponding inclined surfaces on the felloe band and on a wedging ring, which latter is held in place by rim lugs, bolts and nuts.

Two collapsible rims mounted on steel felloes are shown in Figs. 22 and 23, the former being a Firestone and the latter a Hayes design. These rims also are made from special section rolled steel, the Firestone rim having two inclined surfaces on the base, while the Hayes rim has one. In each case the wheel felloe is made with a lip on one flange, affording an inclined surface on which one of the wedge-shaped portions of the rim rests. In the Firestone the other side of the rim is supported by a wedge ring, while in the Hayes it is supported by lugs. Rims with inclined seating surfaces are also made from plain strip steel.

Rims of the split type are made of strip steel and they must be provided with means for locking the ends together after the tire is in place, and with means for driving, that is, for preventing the rim from slipping on the felloe circumferentially under driving and braking torque. Usually the locking and driving means are combined, or at least they are located close together. In Figs. 24 and 25 are shown two forms of locking and driving means for collapsible rims due to the Jaxon Steel Products Company. In the design shown



FIGS. 26 (LEFT) and 27 (UPPER RIGHT).—KELSEY SPLIT-RIM LOCKING AND DRIVING DEVICES.
FIG. 28 (LOWER RIGHT).—RIM WITH SELF-LOCKING SIDE RING.

in Fig. 24 a U-shaped lug is riveted to one end of the rim, and a swinging bar forming a hook is carried by the other end, the outer part of the hook engaging into the slot of the U-lug. In the design shown in Fig. 25 a channel-section locking bar is riveted to one end of the rim and has a slot at its other end which passes over a lug projecting inwardly from the other end of the rim. The lug has a hole in it which takes a split pin. The flanges of the locking bar channel are for driving purposes. They are cut away at the middle of their length to form a slot to admit a bolt passing transversely through the steel felloe.

Two locking and driving devices made by the Kelsey Wheel Company are shown in Figs. 26 and 27. One of them consists of a hinge of which the two plates are riveted to the two ends of the rim respectively. Unlocking is effected by withdrawing the hinge pin. In another design one end of the rim carries a plate with a slot with inclined sides, into which the projecting portion of a plate on the other end of the rim enters, the latter having a form corresponding to that of the slot.

Figure 28 shows a straight-side tire rim with split, self-locking side ring. This is a particularly heavy design much used on taxicabs.

CHAPTER XIX

Chassis Lubrication

All bearings of the chassis at which there is any appreciable motion must be lubricated at intervals, if their rate of wear is not to be excessive. It is fortunate that all or nearly all of the bearings whose shafts rotate at high speed are completely enclosed and can be lubricated by the oil-bath or flood system. On the other hand, there are numerous bearings which are not thus enclosed and must be lubricated either individually or from some central chassis lubricating system. This includes the bearings of the spring shackle bolts, of the brake rigging and of the steering linkage.

The earliest method of lubricating these chassis parts consisted in providing them with individual "screw-down" grease cups (or Stauffer lubricators as they are called in Europe). Some cars were fitted with more than thirty such grease cups, many of them in very inaccessible places, which had to be screwed down every 150 miles or so. If this operation of turning the heads of the grease cups was onerous, that of refilling the cups with grease was even more objectionable, on account of its messiness. Needless to say, as long as this system of lubrication prevailed, cars were very much neglected, and their rate of depreciation was quite high in consequence.

About 1916 a movement set in to replace grease lubrication of chassis parts by oil lubrication. It was argued that oil was a much better lubricant than grease, as the latter, in addition to mineral oil, contains other substances which are added merely for the purpose of giving it "body," and in themselves have no lubricating value.

Three different methods of feed were used, viz., gravity feed, wick feed and pump feed. Gravity feed was the simplest, but involved the disadvantage that it was difficult to regulate the rate of feed. It was necessary to locate the supply at a higher level than the bearing to be lubricated, and in case the bearing was at all loose, the miniature cups

generally provided were quickly emptied, which meant a waste of oil and sloppiness.

Where gravity feed was used the bearing was generally provided with an oil cup having a snap lid. In order to reduce the rate of outflow, these oil cups sometimes were provided with a wad of cotton waste retained in place by a wire screen. Gravity feed from small oil cups as applied to a spring shackle is illustrated in Fig. 1.

Wick Feed—Wick feed from an oil well or magazine formed integral with the bearing to be oiled was probably applied first in connection with the Class B Army truck and is due to Cornelius Myers. Two applications, to a spring-shackle and to a brake-shaft bearing, are shown in Figs. 2 and 3. An advantage claimed for the system is that it feeds only when there is motion at the bearings. The oil reservoirs are located either below the bearing or at the same level, being

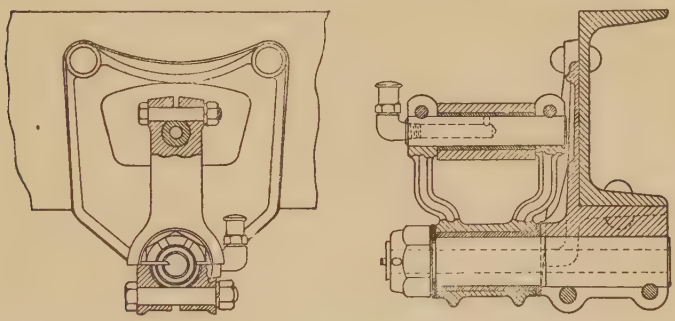


FIG. 1.—OIL CUP LUBRICATION OF SPRING SHACKLE.

cored in brackets, spring shackles or the hubs of levers. One-quarter-inch wicks extend from the bottoms of the reservoirs to the bearing surfaces, and carry oil to them. One of the core holes of the part containing the reservoir serves as the filler hole and is closed by a wing plug. The same method is applied to the lubrication of the front axle steering knuckle pins and thrust washers. The knuckle pins are drilled out and have a wick inserted which raises oil from the reservoir at the bottom of the steering head and feeds it to both bearings and to the thrust washers at the top of the knuckle.

With wick feed the rate of feed is subject to control to a certain extent, as it depends upon the wick material and the tightness of its fit in the hole. Cotton wicking feeds faster than felt wicking, and a loosely fitted wick faster than a tight

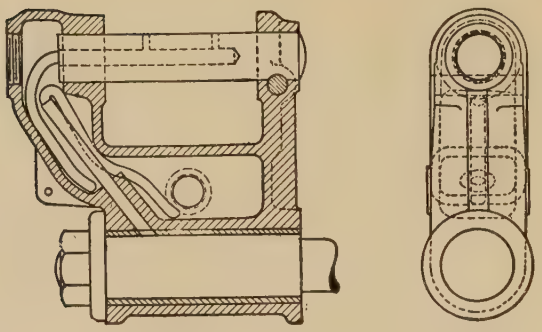


FIG. 2.—WICK-OILING OF SPRING SHACKLE.

one. The rate of feed is increased also by vibration, by an increase in the surface speed of the journal, and by looseness of the bearing. On all bearings carrying a non-reversible load it is advantageous to have the oil hole on the unloaded side. Engine oil is usually employed in connection with wick feed. One of the advantages claimed for the system is that even if dirty oil should be introduced into the reservoirs, it could not reach the bearings.

Pump Feed—An improvement on the original method of oil lubrication by gravity feed was the pump-feed system, which had the advantage that with it there was more pressure on the oil, and the latter was more likely to reach the bearing surfaces where it was needed. Several designs of so-called plunger oilers made by the Madison-Kipp Company are illustrated in Fig. 4. A small oil cup is screwed into the spring

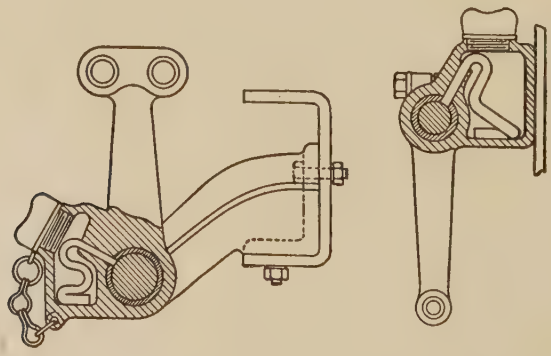


FIG. 3.—WICK-OILING OF BRAKE-SHAFT BEARING.

bolt or shaft from the end. It contains a plunger which normally is held by a coiled spring in its innermost position, but the plunger is fitted with a finger grip by which it can be withdrawn. When thus withdrawn it uncovers an opening in the wall of the oil chamber on top, through which oil can be fed with a squirt can. The plunger is then released and returns to the inward position, forcing the oil through the drilled-out pin to the bearing surface. When no oil is being fed the hole is closed and no dirt can enter.

High-Pressure Gun System—A great improvement in chassis lubrication was introduced in 1923 by the Bassick Manufacturing Company under the name of the Alemite high-pressure system. Use is made of a grease gun with a

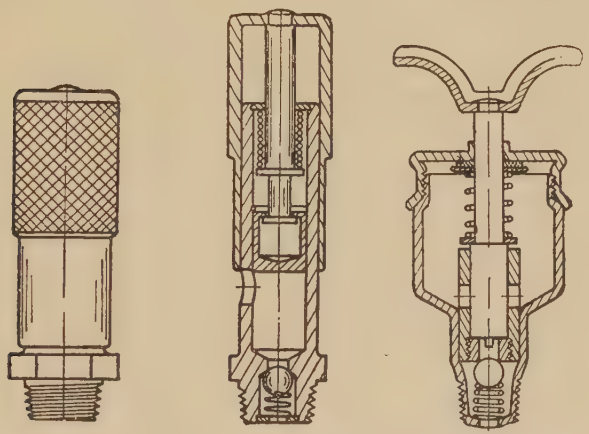


FIG. 4.—PLUNGER OILERS.

piston that is moved forward by a threaded shank or piston rod, by means of which it is possible to create a pressure of several thousand pounds per square inch on the grease. The gun is provided with a length of flexible metal hose, fitted at the free end with a nozzle, which is adapted to make connection with a fitting screwed into the part to be lubricated, by means of a quick-action bayonet joint. Some of the fittings used with the Alemite system are illustrated in Fig. 5. They are made with either a pipe thread or machine screw thread at the inner end, and there is a ball check valve in the other end. The spring for this check valve rests against a cross-pin with which the bayonet-type nozzle engages. Straight fittings are used in positions that are readily accessible with

the gun, while in less accessible positions use is made of angular fittings. Dust caps are provided for the fittings to keep dirt out of the bearings.

Alemite guns or compressors are made in two types. In one the grease is fed by turning a handle at the end of a screw which moves the piston axially in the pump cylinder, and this type is always provided with a flexible hose. In the other, known as the twist compressor, the mere operation of attaching the compressor to the fitting develops pressure on the grease in the nozzle.

Another somewhat similar system is known as the Zerk.

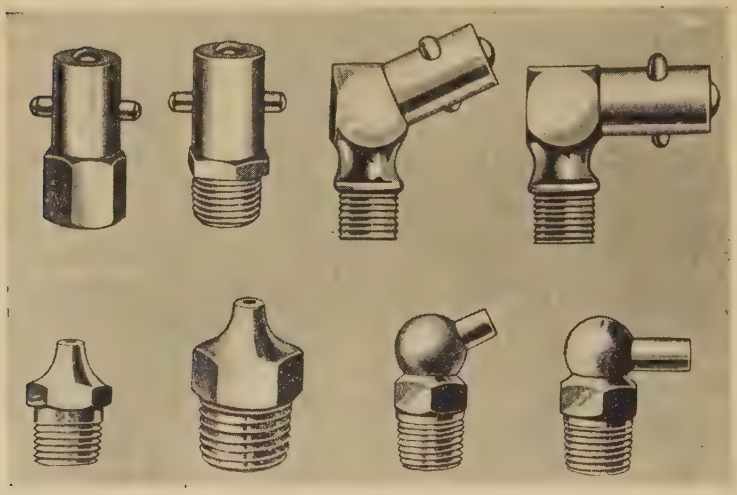


FIG. 5 (TOP).—ALEMITE LUBRICATOR FITTINGS.
FIG. 6 (BOTTOM).—ZERK LUBRICATOR FITTINGS.

Fittings of this system are made with a tip, the outer edge of which makes a line contact with a cupped surface on the compressor nozzle, thus effecting a leak-proof joint. These fittings also contain a ball check valve. In the Zerk compressors the piston is moved directly by means of a pistol-grip secured to a piston-rod passing through the removable head of the compressor. The compressors are made in two types. In one the grease is put under the desired pressure by the piston directly, while in the other use is made of vacuum displacement and atmospheric pressure. In operation, the cup-shaped surface of the nozzle is placed against the tip of the

fitting and the pistol-grip is forced forward as far as it will go, which forces grease into the bearing at pressures up to 5000 lbs. per sq. in. Fittings for the Zerk system are shown in Fig. 6. These high-pressure systems of chassis lubrication are now in very extensive use.

Centralized Systems—It has long been recognized that the ideal system of chassis lubrication is the so-called centralized system, in which all points to be lubricated receive oil from a single central source that is actuated either by the driver at intervals, or automatically through some force having its origin in the motion of the vehicle. The system, of course, must not be too expensive to install; it must feed oil to the different bearings in proportion to their requirements, must cease feeding when the vehicle stops, and must be able to withstand the vibration of the chassis and the stresses due to spring action on the connections leading to the axles. A number of systems have come into use, and as all of them

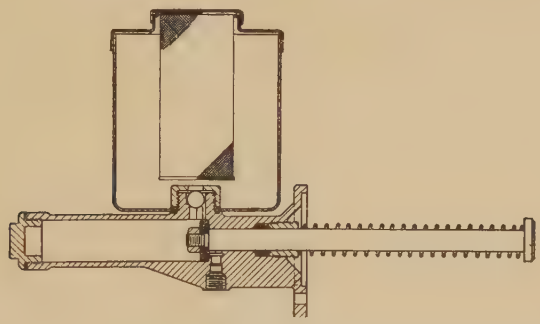


FIG. 7.—RESERVOIR AND PUMP OF BOWEN SYSTEM.

were developed and are being exploited by specialists, perhaps the best way to handle the matter is to describe briefly each of the more prominent systems.

Bowen "One-Shot"—This system comprises a combined reservoir and pump, which may be located either under the floorboards, in such a position that the button at the end of the pump plunger is convenient for the driver to press with his foot, or on the forward side of the dash, with the plunger projecting rearwardly through the dash. Normally a spring surrounding the plunger will hold it in the "out" position, and the pump barrel will fill with oil from the reservoir by gravity. When the operator presses on the pump plunger with his foot, he forces oil into the distributing line and puts

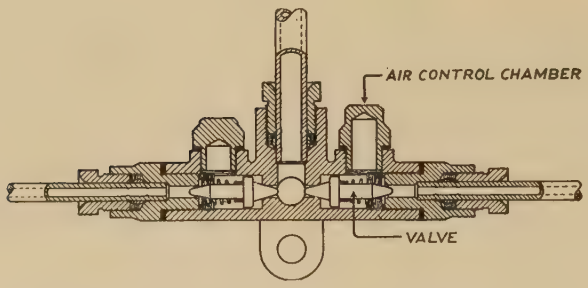


FIG. 8.—CONTROL HEADER OF BOWEN SYSTEM.

it under pressure. A sectional view of the floor-type of reservoir and pump is shown in Fig. 7.

From the pump the oil is forced to so-called control headers, from which latter connection is made directly to the bearings to be lubricated. One of the control headers is shown in section in Fig. 8. There is a double conical valve in the control header for each outlet. When the oil in the line is put under pressure, each valve is forced from the seat to which it is pressed by a light coiled spring, against an outer seat, thereby closing the delivery tube leading to the bearing. While the valve is in this position, oil is forced into a small vertical air chamber above it. As soon as the operator releases the pump plunger, the pressure on the oil in the line drops to nothing, and the valve, under the pressure of the spring, returns to its inner seat. This action

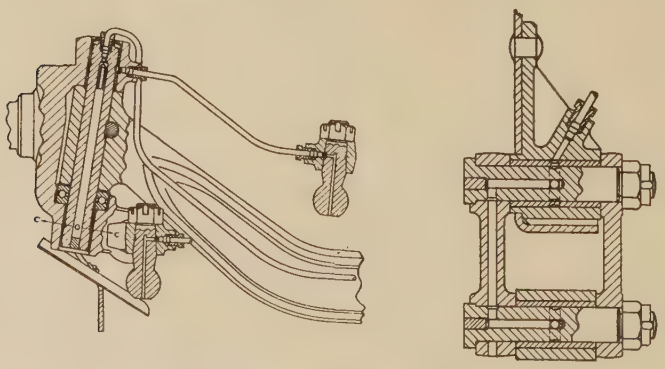


FIG. 9.—CONNECTIONS OF OIL TUBES TO STEERING KNUCKLE BEARINGS AND SHACKLE.

opens the delivery line to the bearing, and at the same time shuts off communication with the distributing line, leaving the oil in the air chamber and delivery line under pressure due to the entrapped air in the top part of the chamber. The size of the air chamber determines the amount of oil which will be fed to the bearing connected to it. A feature of this system is that a break of one or more delivery tubes will not disturb the delivery through any supply line, nor drain the reservoir.

With the Bowen system connections are made by means of heavy-walled copper tubes. Methods of making connection

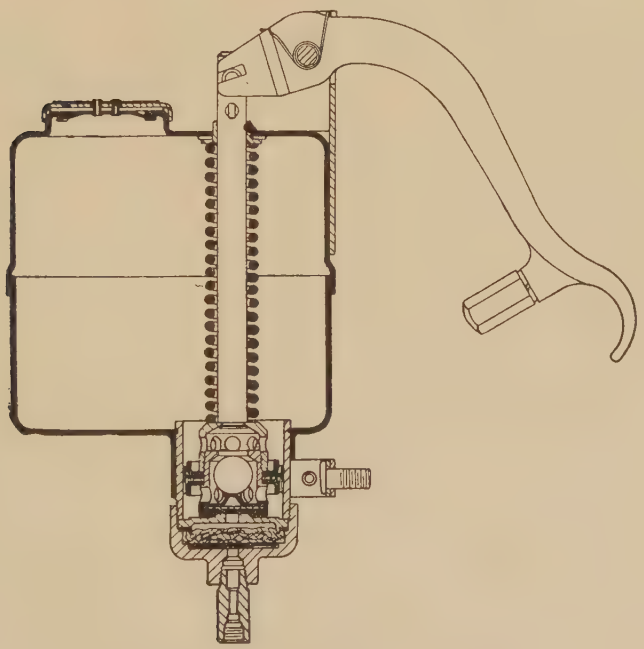


FIG. 10.—BIJUR RESERVOIR AND PUMP.

to all of the points of the chassis have been worked out, and some of them are illustrated in Fig. 9.

Bijur System—The Bijur system also has a combined reservoir and pump which is secured to the forward side of the dash, the reservoir in the latest design being made of two cup-shaped pressings which are soldered together. The pump is really at the bottom of the reservoir and has a leather-cup

piston, within which there is a ball-type of check valve. When the piston is raised against the pressure of the spring surrounding the piston rod, the oil passes through the hollow center of the piston and fills the delivery chamber below it. The spring is of such design that its pressure does not vary much within its range of action, and when pressure on the pump lever is removed by the driver, the spring forces the piston down and puts the oil in the line under pressure. Before leaving the pump unit, the oil is forced through a felt filtering disc at the bottom thereof, to remove any solid particles there may be in it.

Close to each bearing to be oiled there is located a metering valve, which serves to determine the rate of feed to the particular bearing. Normally the metering valve is screwed directly into the shell of the bearing to be oiled. Upon entering the metering valve, the oil is compelled to first flow

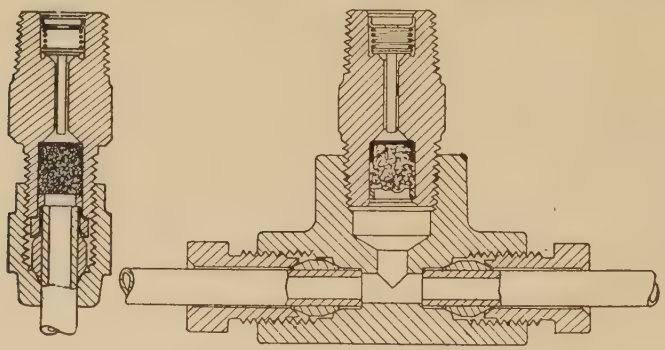


Fig. 11.—Bijur Metering Units.

through a filter composed of a felt plug encased in a gauze collar. The object of this filter is chiefly to catch any dirt or metal chips there may be in the line when first assembled. Directly beyond the filter is the metering valve proper, which consists of a throat in which is located a pin of from 0.001 to 0.010 in. smaller diameter than the throat. These metering valves are made in five capacities, which with medium engine oil at a temperature of 70° F. give deliveries of from less than one drop to 14 drops per minute. At the inner end of the metering valve there is a check valve, which has a facing of varnished silk that is impervious to oil and at the same time is soft enough to give an oil-tight joint with the metal seat under the light pressure of a phosphor bronze spring

by which it is seated. The reservoir with pump is shown in section in Fig. 10.

With this system, as with all centralized chassis lubricating systems, all of the bearings are connected in parallel, and since the pressure which forces oil to the bearings is the same for all, the relative amounts of oil received by the different bearings depend upon the resistance of the passage to and through each bearing. Therefore, if this resistance were due only to the tube and the bearing, and one of the bearings should become very loose through wear, it might get most of the oil delivered by the pump and starve the other bearings. It is to prevent this that the metering valves (Fig. 11) are installed. The passages through these valves are so small that their resistance to oil flow is many times as great as that of even a tight bearing, and the delivery of oil to the bearing therefore depends almost entirely upon the metering valve and is independent of the condition of the bearing.

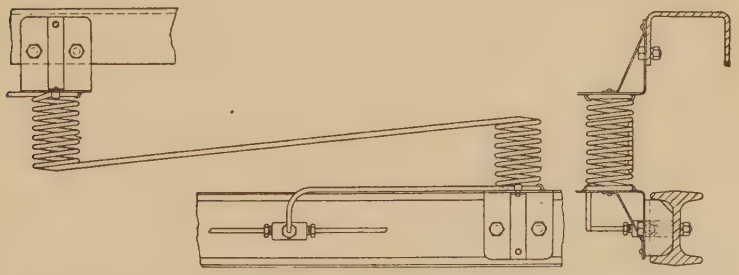


FIG. 12.—METHOD OF CARRYING OIL LEAD FROM FRAME TO AXLE.

Bijur also has worked out methods of connection to all parts of the chassis, and of particular interest is his method of connection from the frame to the axle (Fig. 12) which is designed to minimize the strain on the tubing due to spring action. Heavy-walled brass tubing is used for the connections. The connection tube consists of a straight center section with a helix at each end, connections being made to the frame and the axle at points a considerable distance apart in a horizontal plane. With this form of connection there is evidently no concentration of stress due to the bending action. In the application of such chassis lubrication systems the oil must always be introduced on the free or unloaded side of the bearing.

Alemite Automatic System—An entirely automatic system, with which the operator needs to refill the reservoir only

after perhaps 2000 miles of driving, has been introduced by the Bassick Manufacturing Company as the Alemite automatic system. For feeding the oil to the various points to be lubricated, use is made of a freely suspended inertia weight, which is balanced by a coiled spring and is set in vibration by the vibration of the car.

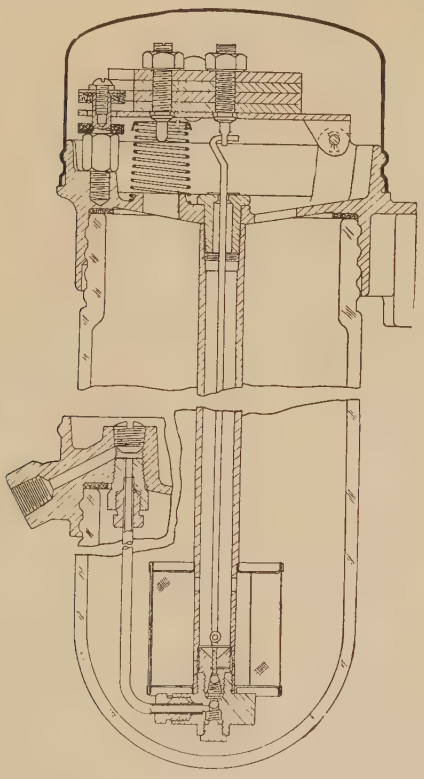
The inertia unit and pumping unit are enclosed in a glass bowl with removable metal cap, as shown in the sectional view, Fig. 13. This glass bowl holds ⅘ quart of oil, sufficient for about 2000 miles' running. Oil is poured in at the top of the unit after the metal cap has been removed. The inertia weight is hinged under the metal cap and balanced on a spring. Such a spring-suspended weight has a constant rate of vibration, independent of the speed of the car. The weight responds by oscillation to the slightest movement of the car, its amplitude of vibration being limited by two stops. A floating plunger rod operating inside a brass tube connects from the inertia weight to the actual pumping unit in the bottom of the glass bowl. To prevent the entrance of solid particles in the oil into the pump, the latter is surrounded with a metal gauze filter of 150 mesh.

When the plunger is near the top of its stroke, oil enters the pump barrel through an inlet port in its wall, and during the down stroke this oil is forced past two ball check valves into the distribution line. Continuous pumping action forces the oil to all of the connected bearings and maintains it under high pressure, the oil entering each of the bearings through a resistance unit. The pump has an exceedingly small rate of delivery, displacing one drop of oil for every forty or fifty strokes. The rate of delivery, morever, can be varied by adjustment. If the screw at the center of the inertia weight (to which the pump plunger is connected) is turned clockwise, the delivery rate is increased, and vice versa. Oil enters the pump through a 150-mesh screen of 6 sq. in. surface area, and since the unit will move only about six gallons of oil during the life of the car, there is no possibility of the screen becoming choked with dirt in the oil.

Figure 14 shows a view of the resistance unit, of which one is connected in the line at each bearing, and serves to regulate the amount of oil received by that bearing. Each resistance unit has a fine passage through it in the form of a helical groove, about 8 in. long, through which the oil must pass. The actual pressure built up in the line depends to a large extent upon the viscosity of the oil, which in turn is dependent upon the temperature under the engine hood.

A wire screen of 150 mesh is inserted in each resistance

unit, chiefly to prevent any dirt which may be in the line when it is assembled, from getting into the helical passage. Each resistance unit also is provided with a check valve, to prevent draining of the system. Some of the bearings of the chassis are located at a higher level than others, and this tends to cause the oil to run out of the lower bearings and draw air into the higher ones, thus draining the distributing



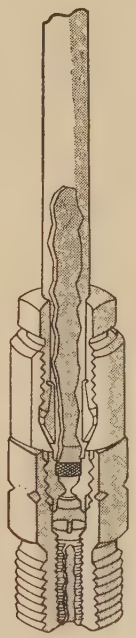


Fig. 13 (left).—Pumping Unit of Alemite Automatic System.
Fig. 14 (right).—Resistance Unit of Alemite System.

line. The problem of producing a check valve that should effect an oil-tight seal under the very slight hydrostatic pressure due to the difference in level of different chassis bearings, was found to be a quite difficult one, but it was solved by the use of a very soft material for the valves.

A number of centralized chassis lubricating systems have been worked out also in Europe. In one, known as the Tecalemit, a vacuum cylinder with piston is mounted on top of the oil pump, and a push-button-type of valve for the vacuum line from the engine inlet manifold to the cylinder is located on the dash. When the operator pushes this button, the engine exhausts the air from the vacuum cylinder, and the piston in this cylinder, being connected to the piston of the oil pump, operates the latter.

In another system, known as the Alcyl, oil is fed from a central reservoir with a number of compartments, by wicks encased in tubes of cotton fabric rendered impervious to the oil by treatment with linseed oil, and encased in a metallic armor. At their ends the tubes are provided with connectors similar in form to cable connectors, which are held in place by bolts that serve as metering devices. With this system the feed is entirely by capillary action.

CHAPTER XX

Gas-Electric Drives

Almost from the beginning of automotive development efforts have been made to combine the flexibility and controlability of the electric motor with the light weight and economy of the gasoline engine, and numerous so-called combination systems have been evolved. Before entering into a discussion of these it may be well to say a few words regarding a form of transmission on which much fruitless effort has been spent by misguided inventors. This includes all forms of transmission known as slip devices.

Slip Transmissions—A friction clutch can be used as a variable speed transmission, for by allowing it to slip at various rates, the speed of the driven shaft can be varied from nothing up to the speed of the driving shaft. Of course, the ordinary clutch, not designed for this purpose, would soon heat and wear excessively, but this difficulty could be overcome to a certain extent by suitable design, using low specific pressures and providing adequate ventilation. However, such a transmission is of no practical use for automobile purposes, for an automobile transmission must not only be capable of reducing the speed of the driven shaft, but also of multiplying the torque of that shaft. This the slip device will not do; it impresses the same torque on the driven shaft as it receives from the driving shaft, and the loss in the transmission therefore is directly proportional to the loss in speed, that is, to the slip. For instance, if the driven shaft turns at only half the speed of the driving shaft, exactly 50 per cent of the power delivered to the slip device is lost in friction at the sliding surfaces.

Slip devices can be made also in the hydraulic and the electro-magnetic forms, and these have the same faults and limitations as the mechanical type. For instance, a device on the driving shaft may be used to set a body of liquid into motion, and the liquid may impinge on vanes carried by the driven shaft. The impact of the fluid on the vanes will tend

to cause the driven shaft to revolve, and if the resistance to motion is less than the force of impact, it will revolve. The speed of motion of the driven part, however, will be less than that of the driving part, and the power loss will be proportional to the loss in speed, as there will be no increase in torque.

An electro-magnetic slip device is represented by an electric motor of the direct-current type. The field frame of such a motor can be developed as an engine flywheel, and if the armature is short-circuited or closed through a small resistance, when the field is revolved by the engine, a current will flow in the armature, and there will be a heavy drag on the armature, which will cause it to revolve if the resistance opposed to its revolution is not too great. In this case, again, the power loss will be directly proportional to the slip. The armature will always revolve slower than the field, and if its speed is only half that of the field, for instance, then one half of the power required for turning the field will be wasted in heating of the armature coils by the current flowing through them. But if, instead of short-circuiting the armature upon itself or closing its circuit through a rheostat or other resistance, the current generated is sent through the armature and field of an electric motor, both a reduction in speed and an increase in torque may be obtained. One of the first sucessful combination systems was based on this principle.

The Entz System—This system, developed by Justus B. Entz, was used for a number of years in the Owen magnetic car, which was equipped with a gasoline engine that was direct-connected to a generator, the field frame of the generator serving as flywheel of the engine. The armature of the generator was mounted on a shaft that was piloted at its forward end in the end of the crankshaft, and at its rear end had a bearing in the housing of the electrical transmission unit. Thus both the field frame and the armature of the generator were capable of rotary motion, the field frame always rotating at engine speed, and the armature at propeller shaft speed, the latter being connected to the propeller shaft at its rear end through a universal joint. The armature always turned at a lower speed than the field frame, and in consequence of the relative motion an electro-motive force was induced in the generator armature windings which was proportional to the difference in the angular speeds of the field and armature, or to the "slip" of the armature. Current was collected from the armature by means of commutator brushes revolving with the field.

This current was sent into an electric motor mounted

concentric with and directly back of the generator, the field of the motor being stationary, and its armature carried by an extension of the generator shaft, to which it was fixed. For high speed, or what corresponded to the direct drive, the motor was entirely disconnected from the generator electrically, and did not play any part in the transmission. The generator armature was then short-circuited upon itself. To produce a certain torque, it was necessary for the generator armature to carry a certain definite curent, and as this current had to overcome only the resistance of the short-circuited armature, little electro-motive force was required to produce it, and it required little slippage of the armature relative to the field frame to generate this electro-motive force. Consequently, the armature turned at nearly engine speed.

To increase or multiply the engine torque, the generator was electrically connected by a controller to the electric motor. In that case, not only was the resistance in the circuit of the generator armature increased, but the generator armature also had to overcome the counter-electro-motive force generated in the motor armature; hence, to produce a given current flow (and a given drag of the generator field on the armature), a much higher electro-motive force had to be produced in the generator armature, which necessitated a much greater slippage. The current sent by the generator into the motor would produce a torque on the motor armature, which torque was added to that due to the drag of the generator field on the generator armature. Hence the torque on the propeller shaft would be increased over that on the engine crankshaft; at a slippage of 50 per cent the propeller shaft torque would be nearly twice the crankshaft torque, and the internal losses therefore were moderate.

Various other speed reductions and torque multiplications were obtained by changing the field connections.

Bus Transmissions—The electric system of transmission seems to have a much better field in city bus work than in private cars, for the latter can be and usually are provided with comparatively very powerful engines, so that nearly all driving can be done on high, and the faults of the sliding change-speed gear are not apparent. What was probably the first successful electric drive for commercial vehicles was developed by the firm of Tilling-Stevens of Maidstone, England, about 1917. Later several American firms developed transmissions of this type and since 1925 they have been extensively used for city and suburban buses. Electric drives of a very similar type are used also for railcars, and it now appears that the electric drive will practically control this

field in the future, for the call is for constantly more powerful engines, and the greater the power to be transmitted the greater the difficulties with clutches and geared transmissions.

An electric-drive system comprising a generator of the separately-excited, shunt-wound or compound-wound type and a motor with series field winding, constitutes really an automatic torque converter. The generator is direct-connected to the engine, while the motor is direct-geared to the vehicle; generator and motor are connected only through electric cables, and there is no direct mechanical connection from the engine to the driving wheels, which tends to smoothen the drive and protect the engine against shock. The electrical connections for the generator of a gas-electric drive are shown in Fig. 1.

Relation Between Current and Torque, Voltage and Speed —When the vehicle encounters increased resistance to motion, as in ascending a grade, the motor is pulled down in speed, develops less counter-electro-motive force and draws more current. Since the current can come only from the generator, the output of the latter will be similarly increased. In fact, the current received by the motor is exactly the same as that delivered by the generator, except for the small quantities required for the generator shunt field and for charging the storage battery.

If the field strength were constant, the generator voltage would be directly proportional to the armature speed, and the generator current to the torque impressed on the generator armature in excess of that necessary to overcome bearing and brush friction. But the field strength is not constant in either the generator or the motor.

The motor has a series field winding, and if its armature current increases, the current through its field coils increases equally. Therefore, since motor torque is proportional to both armature current and strength of magnetic field, it always increases faster than the armature current, and a curve of motor torque with respect to current flow is convex toward the current axis over the range of currents corresponding to normal loads.

The exact opposite holds in the case of the generator. Field excitation of the generator is derived principally from a shunt winding connected across the generator mains. When the current delivered by the generator increases there is an increase in potential drop in the armature windings, and consequently a decrease in terminal voltage and in field excitation due to the shunt coils. Hence, even if the generator speed remained absolutely constant, the field strength would

decrease with an increase in armature current. However, with the increased torque load on the engine due to the greater armature current, the engine speed will drop, which results in a further decrease in terminal voltage and in field strength. Finally, the so-called armature reaction—the magnetic effect of the armature current—causes a shift in the direction of magnetic flow through the armature core, and a consequent decrease in terminal voltage.

Thus there is an inherent tendency for the field strength of the generator to decrease with an increase in armature current. The horse-power output of the generator is proportional to the product of the voltage and the current, and if the generator voltage drops while the current increases, there is a tendency for the engine load to remain constant irrespective of the rear axle torque (and motor torque). If

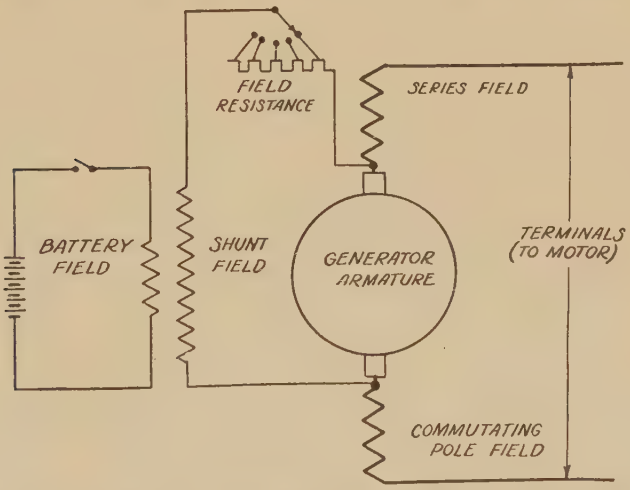


Fig. 1.—Diagram of Generator Connections (Westinghouse).

this tendency is not sufficiently pronounced it can be strengthened by providing the machine with a small reverse series winding, which tends to demagnetize the field.

This automatic change of torque multiplication by the electric system is well illustrated in Fig. 2, which represents torque curves of generator input (engine torque) and of motor output. It will be noticed that with a current flow of 130 amperes the generator torque is about 230 lb.-ft and that there is only little variation in the generator input torque

from this point on. On the other hand, the motor torque, which is 74 lb.-ft. with a current flow of 130 amperes, reaches 432 lb.-ft. for a current flow of 400 amperes. If we consider 75 lb.-ft. the minimum motor torque with which the vehicle can be moved, the range of torque variation in this case is

$$\frac{432}{235} \times \frac{205}{75} = 5 \text{ to } 1 \text{ (approx.)}.$$

The variation in the speeds of the generator and motor in relation to current flow is shown in Fig. 3. It will be seen

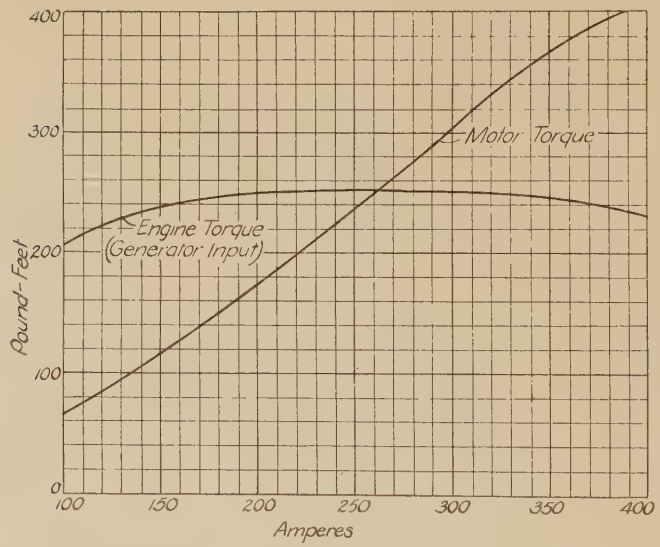


FIG. 2.—CHANGE OF GENERATOR INPUT TORQUE AND OF MOTOR TORQUE WITH CHANGE IN ARMATURE CURRENT

that the generator speed is nearly constant over a wide range in current flow, from which it follows that with electric drive the engine runs at nearly constant speed. Moreover, with both the speed and the torque of the engine nearly constant, the engine output is practically constant under all operating conditions. The motor speed, on the other hand, varies inversely as the motor torque, high motor speed corresponding to a small current and to low motor torque, and low motor speed to high motor torque.

Separate Excitation—When it was first attempted to connect a vehicle motor directly to a generator driven by an internal combustion engine without having a battery floating on the line, one difficulty that was experienced was that when the driver opened the throttle quickly for a rapid get-away, the engine would be momentarily without adequate load and would "race," which not only was injurious to its mechanism but resulted in unpleasant vibration. The reason for this was that the generator had to build up its field from its small amount of permanent magnetism. "Picking up" of the generator under such conditions is a very much slower process than the acceleration of the engine by opening the throttle

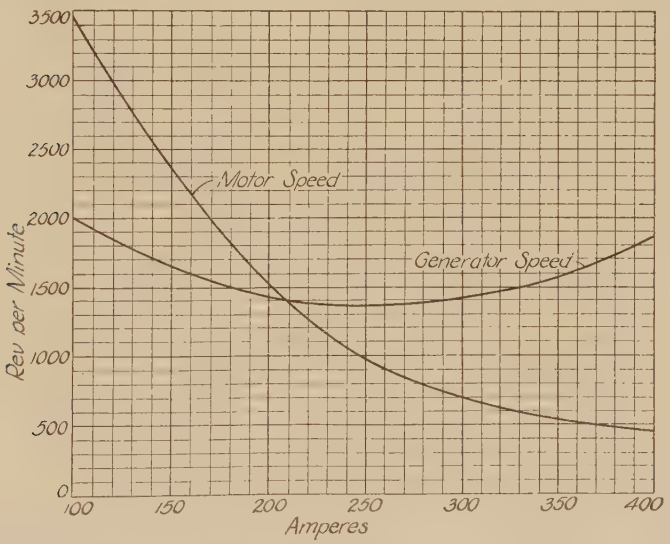


FIG. 3.—CHANGE OF GENERATOR AND MOTOR SPEED WITH ARMATURE CURRENT.

wide. The permanent magnetism in the fields of automobile generators is likely to be particularly small, for the reason that magnetic materials of very high permeability have to be used, to reduce the weight to a minimum. This calls for specially soft iron or steel, which has a very low magnetic retentivity.

This difficulty was overcome by providing a certain amount of permanent or separate excitation, the current for which, of course, must come from a battery. In one case,

in order to save on battery current, a small exciting generator was provided whose field was excited with battery current, while the current generated by it was used to excite the field of the main generator. This scheme seems to have been given up again, however, probably because it involved unnecessary complication.

At the present time the generators are provided with an auxiliary field winding supplied with current from the lighting battery, which produces a magnetic field of sufficient strength so the generator can pick up promptly as soon as the throttle is opened. The greater part of the magnetizing force comes from a shunt coil, however. In fact, in the G. E. system, the circuit diagram of which is shown in Fig. 4, the auxiliary exciting coil or teaser coil is connected in series

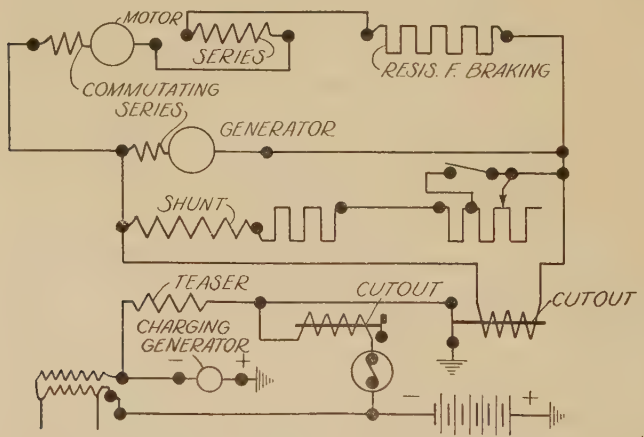


Fig. 4.—Circuit Diagram of General Electric System when Braking.

with an automatic cut-out, and the circuit is opened as soon as the voltage of the generator reaches a certain value.

Commutating Poles—In addition to the teaser field there is a third field winding on small commutating poles located between the main field poles. These commutating poles serve to make possible sparkless commutation under all load conditions. In an electric generator the ends of each armature coil (or of a series of coils) are connected to adjacent commutator bars. For an instant while both of the bars to which the coil is connected are under a brush, the coil is short-circuited, and if any voltage at all is being induced in the short-

circuited coil, a current will flow in it. In order to insure sparkless commutation, this current should be of the same magnitude and flow in the same direction as the current which must flow through the coil after the short circuit is broken. This means that the current and induced voltage of the short-circuited coil must vary with the load on the generator. Hence, commutating poles are so located that the armature coils are under their influence while short-circuited, and these poles are provided with series field windings (see Fig. 1) so that their strength varies with the load on the armature.

Single Motor vs. Two Motors—In connection with the motors, the question arises whether one or two should be used. One motor evidently has the advantage from the points of view of first cost, weight and simplicity. There are, however, a number of advantages connected with the double-motor system. Double motors may be applied in two different ways.

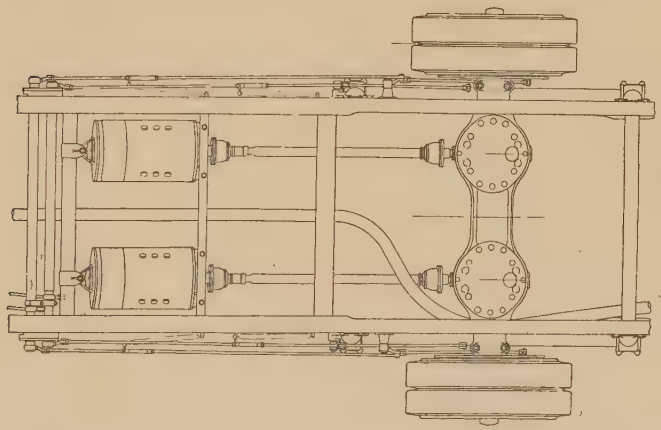


Fig. 5.—Two-Motor Drive, Each Motor Independently Geared.

Each motor may be independently geared to one of the driving wheels (Fig. 5) and the differential gear may then be dispensed with. In this case, however, two bevel or worm-gear reductions are required, and the axle has to be made of special design, which detracts from the advantages. The other method consists in coupling two motors end to end and driving through a single propeller shaft and a differential, as shown in Fig. 6.

One advantage of the double-motor drive is that if one motor should become defective, the bus could complete its

route on the other one, thus obviating the need for towing due to motor defects. But since generator and engine defects, etc., would still call for towing, the degree of reliability would not be greatly increased. The chief advantage of double motors undoubtedly resides in the possibility of connecting them to the generator either in series or in parallel. When connected in series, each motor will have only one-half of the generator voltage impressed upon its terminals, and will therefore run at a low speed. On the other hand, each motor will then take the full current of the generator and therefore produce greater torque as if it were connected in parallel with the other and received only half of the generator current. The motors may not be capable of taking the whole of the generator current continuously without heating, but for short periods of heavy pulling additional torque can be obtained in this way.

Weight of Electric Drive—One serious objection to the electric transmission is the great weight of its components, and designers naturally are bending their efforts toward reducing this weight. Taking the generator and motor or motors alone, the weight varies from about 24 lbs. per engine hp. for the smaller sizes to about 18 lbs. for the larger sizes. For the sake of comparison it may be pointed out that modern six-cylinder bus engines weigh only about 12 lbs. per horse power. The electrical system, of course, replaces the clutch and transmission and thus eliminates the weight of these parts, but this saving is probably wiped out by the additional frame weight required to support the heavy electrical units. It may be possible to save a fraction of the flywheel weight, as the generator armature has considerable inertia and, besides, the engine does not have to operate at such low speeds as with mechanical drive, and therefore does not need so much flywheel capacity.

The capacity of electrical machinery in general is limited by heating. There are two principal kinds of losses in electric machines which determine this heating—the copper losses (or C^2R losses) and the iron losses (or hysteresis and eddy-current losses). As with other power machinery, in order to get the most power from a machine of given dimensions (and, therefore, of a certain weight) it is necessary to operate at high speeds. Generator rotative speed is limited by engine speed, but even with this restriction something can be gained by using armatures of fairly large diameter and increasing the number of poles. The strain on armature banding, however, limits the armature diameter.

Reducing Internal Losses—The greatest current draw occurs during acceleration. Motor buses in city service, where frequent stops have to be made, must be capable of a fairly high rate of acceleration in order that a good schedule may be maintained. In fact, more rapid acceleration is one of the chief claims put forward in connection with the electric drive. The acceleration need not be as high as that of high-powered passenger cars, but a maximum acceleration of 2 m.p.h.p.s. should be attainable with a moderate passenger load. To make it possible to obtain the high current necessary for such acceleration with generators and motors of moderate weight, it is necessary to keep down the armature C^2R losses. In other words, the armatures must be designed to have a low resistance.

To obtain this low resistance a large amount of copper

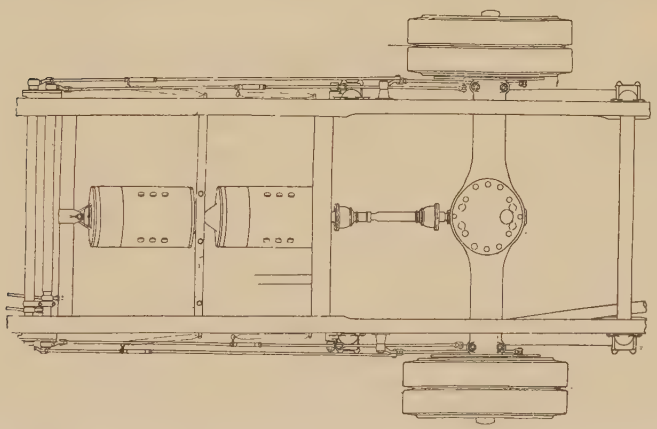


Fig. 6.—Two-Motor Drive with Motors Arranged in Tandem.

must be placed on the armature, and that can be done only by reducing the amount of iron in the core. This involves higher magnetic densities, which tend to increase the iron losses. These latter losses, however, are a function of the speed, and during periods of acceleration the motor speed at least is below normal, so these losses are not so serious as the copper losses. If an electric drive were specially designed for use on long distance buses, in which both generators and motors would run at their maximum speed most of the time, it would evidently be advantageous to reduce the armature

losses under these conditions by using more iron and less copper in the armature.

Ventilation—In addition to making efforts to keep down the losses, and consequently the heat generation, designers of electric transmissions provide means for getting rid of the heat as quickly as possible. It is naturally rather difficult for the armature of a completely enclosed electric machine to dispose of the heat generated in it, and a ventilating system is therefore provided.

Figure 7 illustrates how the ventilating system is arranged on the Westinghouse machines. A centrifugal fan is mounted on the driving end of the armature, and expels the heated air from the housing of the machine through a circular series of openings therein. There are two sets of air inlets and two passages for the air through the machine. Air entering through a large screened opening at the bottom of the housing at the commutator end passes through the spaces between field

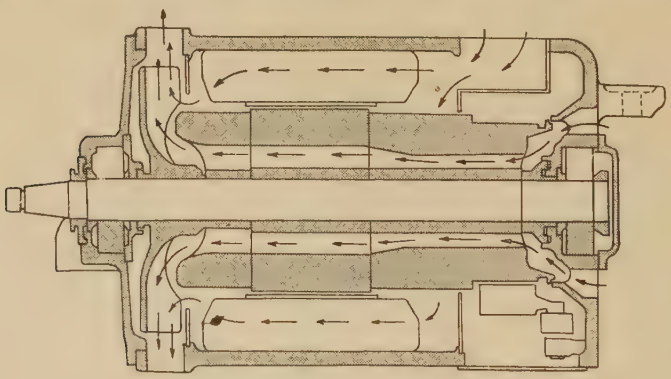


FIG. 7.—DIAGRAM OF GENERATOR AND MOTOR VENTILATION (Westinghouse).

poles to the driving end, and particularly cools the field coils. A shield is placed over the commutator, so that this air, which is likely to carry dust—and in wet weather moisture from the wheel splash—cannot come in contact with the commutator and brushes. There is another series of air inlets in the housing around the bearing at the commutator end, and air entering through these openings passes through a spider supporting the armature core on the shaft. Finally, to prevent injury to the machines from excessive heating in very severe service, the insulating materials used are chiefly such as are not affected by heat; that is, mica, asbestos, etc.

Efficiency—With a mechanical transmission, when most of the driving can be done on the direct drive, the transmission losses amount to only a few per cent, and even in the lower gears the efficiency is between 85 and 95 per cent. When the energy has to be converted from the mechanical to the electric form and then back again, the losses naturally are considerably greater. A curve giving the overall efficiency for a certain generator and motor combination is shown in Fig. 8. It will be noticed that the efficiency varies between 65 and 75 per cent for the greater part of the range of current flow, while for very large currents it drops below 50 per cent.

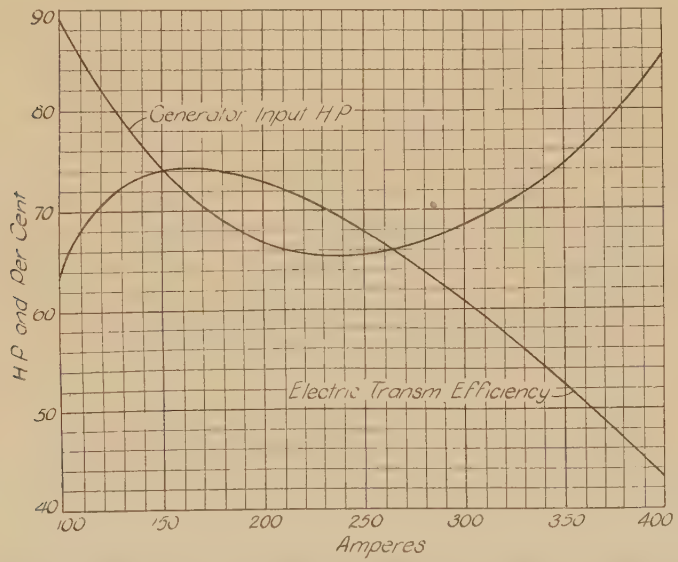


FIG. 8.—VARIATION OF GENERATOR INPUT AND OVERALL ELECTRICAL EFFICIENCY WITH ARMATURE CURRENT.

There is therefore a decided difference in favor of the mechanical drive from this point of view. But in bus operation, the same as in connection with private cars, fuel cost is only a small fraction of the total operating expense, and if the public appreciates the smoother acceleration of the electrically driven bus, the slight extra fuel cost will not stand in the way of the adoption of the drive. It has been pointed out, moreover, that a comparison of the efficiencies of the two

systems as determined on the test stand does not give a correct view of the actual efficiencies. The best plan is to compare the fuel consumptions of machines of equal carrying capacities when operating over similar routes, and it has been found that under these conditions the bus with electric drive consumes about 20 per cent more fuel.

Electric Braking—When there is no direct mechanical connection between the engine and the driving wheels, the engine cannot be used as a brake. The electric motor, however, lends itself to the same purpose. It can be used either as a mild brake, to prevent the bus from attaining too high a speed in descending grades, or as a severe brake, for emergencies. In the first case the motor is merely disconnected from the generator and its circuit is closed through a resistance. The braking power obtained depends on the resistance in circuit and may be made variable if desired, the same as the braking power of an engine can be varied by means of the

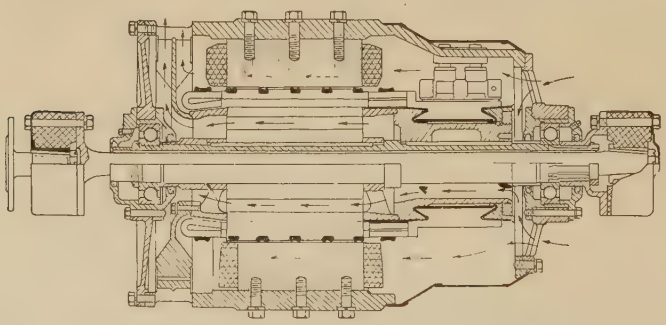


Fig. 9.—Flexible Driving Connection for Generator (G. E.).

throttle. Ordinarily, however, only one step is provided. Emergency braking is effected by connecting the motor to the generator in reverse, with a resistance in circuit. The braking effect then depends on the throttle position, and can be controlled by means of the throttle. Connections for service braking in the General Electric system are illustrated in Fig. 4.

Accelerator Switch—The generator and motor of an electric transmission system must be designed to fit the engine with which they are to be used. The engine operates without governor, and attains a speed which is determined by the load on the generator. The electrical system is so designed that the generator over the greater part of the operating

range loads the engine to about 85 per cent capacity, based on bench tests of the new machine, which leaves a satisfactory margin for loss of power due to any of the various possible causes. The speed to which the engine is held down ordinarily is not that at which it develops its maximum power but a considerably lower speed, as if the engine were habitually operated at the peaking speed, its life would be short. However, occasionally it is desirable to operate the engine at or

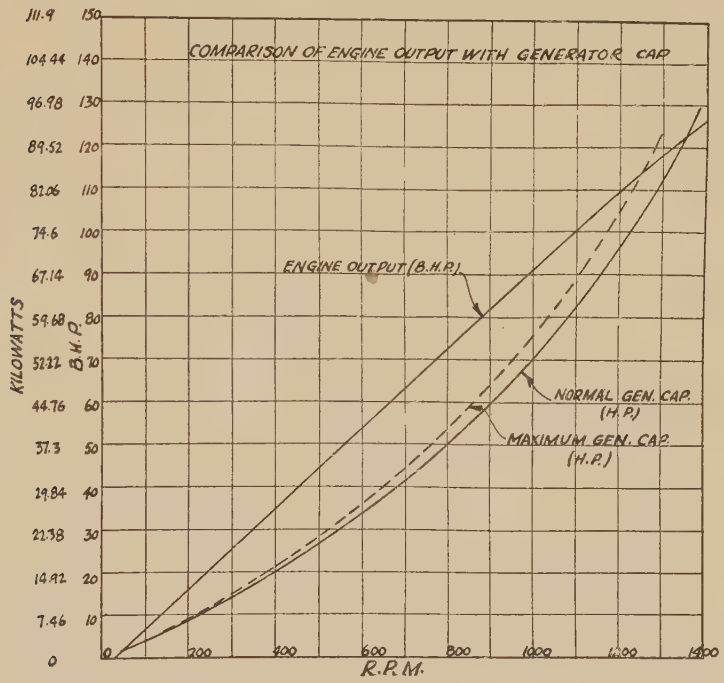


FIG. 10.—VARIATION OF ENGINE AND GENERATOR CAPACITIES WITH SPEED.

near its peaking speed, and a simple means of accomplishing this is available with the electric transmission system. All that is necessary is that the engine torque load be reduced, which can be accomplished by reducing the generator field strength, since the driving torque of the generator is proportional to the armature current and the field strength. Therefore, a resistance for the shunt field circuit of the generator is provided which is ordinarily cut out by means of a switch

(see Fig. 4). This switch is controlled by a pedal, and when it is desired to speed up the engine and secure the advantage of the additional power which it will generate at the higher speed, the driver need only press down on the pedal, thereby including the resistance in the field circuit.

Long experience with direct-connected generators has shown it to be desirable to have a small degree of flexibility in the driving connection between the engine shaft and the armature shaft. This flexibility can be obtained in an automotive generating set by making the armature shaft hollow and driving through a shaft of high-tensile steel and comparatively small diameter, which connects to the crankshaft and to the armature shaft (at the far end) by flexible couplings. With this arrangement it is unnecessary to have the engine and generator in perfect alignment. Such a drive as worked out by the General Electric Co., is illustrated in Fig. 9. This illustration also shows another method of armature ventilation.

Proportion of Generator to Engine Capacity—If an electric drive is to give satisfactory service, the capacities of its components must be carefully balanced against the output of the engine. In Fig. 10 are given graphs of maximum output in relation to speed for both a gasoline engine and a shunt-wound generator. The engine curve shows the maximum power which the engine is capable of developing at any particular speed, while the generator curve shows the maximum power which the generator can absorb indefinitely at any given speed without overheating. It will be seen that if engine and generator capacities are equal at the peaking speed of the engine, the capacity of the generator will be below that of the engine at all lower speeds, and if the engine is operated under full throttle at comparatively low speeds for any length of time, the generator will overheat. There is more chance of trouble from this cause in the case of railcars than in that of motor buses, because of the much greater proportion of weight to power in the former, which means longer periods of acceleration and generally a higher load factor. Such trouble can be prevented by using comparatively large generators.

Since the electric drive comes higher in first cost than a mechanical drive and leads to increased fuel consumption per bus-mile, the logic of its use is not apparent at first sight, but experience has shown that on account of the continuous, uninterrupted acceleration it enables a bus to cover more miles per day, it decreases the driver's duties, thereby decreasing accidents and making the driver's job more attrac-

tive, and it results in smoother operation, thereby attracting patronage and increasing the life of the engine and the bus as a whole. Finally, it has the advantage of leaving the designer greater freedom in the arrangement of his chassis, as the engine need not be mechanically connected to the driving axle. In some buses with electric drive the power plant is placed across the chassis at the rear. Comparative acceleration curves of buses with mechanical and electric drives are shown in Fig. 11.

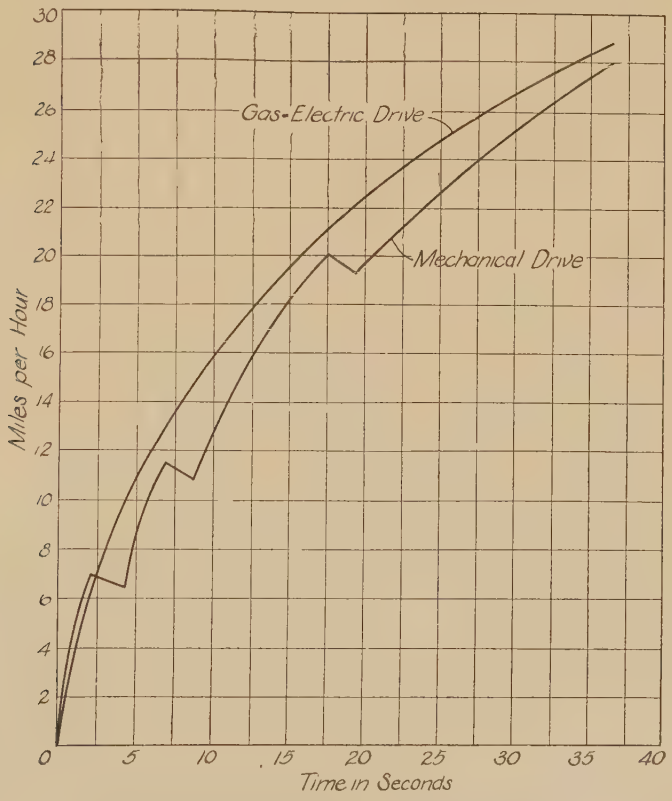


FIG. 11.—COMPARATIVE CURVES OF ACCELERATION OF MACK BUSES WITH MECHANICAL AND ELECTRIC DRIVE.

(The mechanically-driven bus weighed 18,000, the electrically-driven one 20,000 lbs.).

CHAPTER XXI

The Chain Drive

Transmission by means of chains and sprockets is still used to a certain extent in motor trucks and tractors, although it no longer enjoys the popularity it did in the early years of the industry. The chain possesses the advantage of a slightly greater flexibility than the shaft drive, hence it tends to protect the engine and tires from shock due to too rapid engagement of the clutch, road shocks, etc. When kept clean, oiled and properly adjusted, the chain is a very efficient means of power transmission. With the constant increase in the proportion of dustless roads the difficulty of keeping it fairly clean is disappearing. One of the largest manufacturers of commercial vehicles in the United States (Mack) has always favored the chain, and another (International Harvester) returned to the chain drive recently after having produced shaft-driven trucks exclusively for a great many years.

If the chain could be enclosed in an oil-tight case and run in oil, it would be a most efficient and reliable means of transmission, but the problem of a light chain case which shall hold oil, not leak and permit of ready inspection of the chains and of adjustment of their tension is a very difficult problem that has never been satisfactorily solved. However, important improvements have been made in the design of chains and sprockets in recent years, especially in the tooth form of the sprockets, so that the efficiency of the chain does not fall off rapidly and the noise of operation does not increase materially as the chain wears and stretches.

Construction of Chains—The only type of chain used in automotive drives is the roller chain, which is one form of the general class of machine-made chains. The chain (Fig. 1) consists of two sets of links, inner and outer, respectively, each set of one kind being joined to two sets of the other kind by means of a bushing and a rivet for each joint. The bushing serves to hold the pair of inner links the proper distance apart, and the rivet has both of the outer links

riveted to it. In passing over a sprocket the rivet turns inside the bushing through an angle which is equal to 360° divided by the number of teeth in the sprocket, first in one direction and then, as it leaves the sprocket, in the other. It is this motion of the joints which is responsible for the wear of chains, and as the motion is less the greater the number of teeth in the sprocket, the advantage of using large sprockets is obvious.

The bushing is surrounded by a roller which contacts with the sprocket teeth. Hence the contact between the chain and the sprocket is a rolling contact, and the sliding takes place between the bushing and roller and between the bushing and pin. The rivets are generally made of nickel steel and the bushings and rollers are hardened.

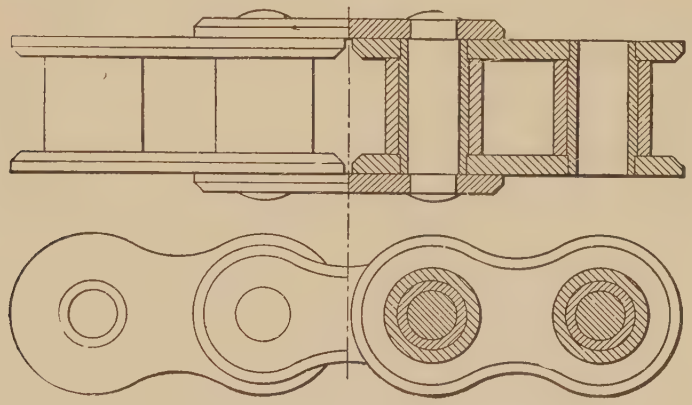


FIG. 1.—ROLLER CHAIN.

Capacity of Roller Chains—The permissible working tension of a roller chain increases substantially as the square of the pitch, because both the pin diameter and the bushing width increase with the pitch, and the product of these two factors is the joint area, to which the working load must be proportional. Hence, from the standpoint of strength it is advantageous to use a large pitch. On the other hand, since the diameter of the sprocket is limited by considerations of ground clearance required, the number of teeth in the sprocket is inversely proportional to the pitch, and since the angular motion at the joints is inversely proportional to the number of sprocket teeth it is directly proportional to the pitch. Hence, by increasing the pitch we reduce the unit bearing

pressure, but increase the motion at the bearings. A chain of smaller pitch operates more quietly and has a longer length of life provided the tension on it is not too high. Roller chains for automobiles are commercially made in pitches varying from ¾ inch to 2 inches, in ¼ inch gradations, and each pitch is made in several different widths of rolls.

The Diamond Chain and Manufacturing Company recommend the following sizes of chains for commercial vehicles of different capacities:

Tons.	Pitch, Inches.	Width, Inches.	Roller Diameter, Inches.
1	1¼	¾	¾
1½	1¼	¾	¾
2	1½	1	⅞
2½ and 3	1½	1	⅞
4	1¾	1	1
5 and over	2	1¼	1⅛

The chain sizes recommended correspond to the S.A.E. heavy series of roller chains (see Appendix). There is now a pronounced tendency toward higher truck speeds, and although adequate strength is vital, if superfluous strength is obtained at the expense of additional weight it is detrimental to performance. As with other mechanisms, as speeds are increased, unit weights must be decreased, and it may be that if the chain should find application on high speed trucks, the medium series would be preferred.

In laying out a chain drive for a commercial vehicle the aim should be to make the chain speed as high as possible, because in any case the average chain speed will be moderate and the higher the chain speed the less the tension in the chain for a given horse power transmitted. The large sprocket wheel must clear the ground by about 7 in. hence the sprocket pitch diameter must be from 15 to 16 in. less than the wheel diameter. The pitch diameter of the front sprocket then depends upon the gear reduction desired. In commercial vehicles fitted with solid tires the total reduction ratio between engine and rear wheel is generally between 6 and 9. It is customary to make the two reductions, at the bevel gear set on the countershaft and at the chains, about equal; hence the speed reduction by the chains and sprockets will vary between 2½ and 3. This gives a front sprocket with the necessary number of teeth to insure proper operation. Sprockets with less than ten teeth quickly destroy the chains. Sprockets with 12 to 13 teeth give tolerably satisfactory service, while sprockets with 15 teeth or more are most satisfactory.

Chain and Sprocket Calculations—The pitch diameter of a sprocket for roller chains may be found by means of the following equation:

$$D_p = \frac{P}{\sin \frac{180°}{N}}$$

where D_p is the pitch diameter; P, the pitch of the chain, and N the number of teeth in the sprocket.

The distance between centers of sprockets for a certain number of links in the chain may be found by means of the equation

$$L = P \frac{Z - \frac{N}{2} - \frac{n}{2} - (N - n)\frac{\beta}{180}}{2 \cos \beta}$$

where Z is the number of links in the chain; N, the number of teeth in the large sprocket; n, the number of teeth in the small sprocket, and β, the angle made by the chain with the line of

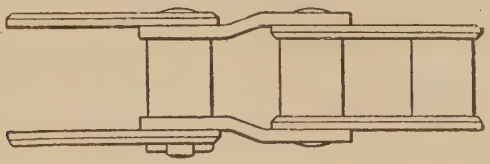


FIG. 2.—OFFSET LINK.

centers. This equation gives the distance required for the chain to run tight on the sprockets. A slight amount of slack is necessary and means for adjusting the distance between centers are generally provided. The distance between centers of sprockets must not be less than one and one-half times the pitch diameter of the large sprocket. Too long chains are also objectionable because of the whipping effect if the chain is at all loose. In commercial vehicle practice the distance between sprocket centers is generally made equal to about twice the pitch diameter of the large sprocket. In fixing this distance it is preferable to figure on an even number of links, because although the use of so-called offset links (Fig. 2) permits of an odd number, this practice is objectionable.

Design of Sprocket Wheels—Sprocket wheels are made from steel plate, drop forgings or, in some instances, cast steel. The larger sprockets are practically always in the form of flat rings which are generally bolted to the brake drums. Front sprockets, on account of their smaller size, wear faster

than rear sprockets and should be deeply case hardened. Front sprockets also are sometimes bolted to separate hubs, the advantages of this construction being that when a sprocket is worn out only the steel disc need be renewed, and that a lot of sprockets can be forced over a mandrel and cut at the same time. Referring to Fig. 3, the width B of the sprocket is made equal to $0.93\ A - 0.006$ in., the tolerance on this dimension being $\pm\ (0.01\ A + 0.002$ in.), and the sprocket teeth are rounded off on both sides to radius $R = 0.43\ P$, P being the pitch of the chain. The center of the arc R lies on the pitch circle (normally $0.3\ P$ below the top of the tooth). The clearance between the side links and the full-width portion of the sprocket $(A - B)$ is made $0.07\ A + 0.006$ in., and the width C at the top of the teeth is $(0.93\ A - 0.25\ P - 0.006$ in.).

Sprocket wheels are cut by means of formed cutters which have been standardized by the Sectional Committee for the Standardization of Gears of the American Standards Association, which also standardized chains and sprocket forms. Fig. 4 shows one of two standardized roller-chain-sprocket tooth forms, the dimensions for different pitches P, roller diameters D and numbers of teeth T being calculated as follows:

$$C = \frac{1.005D + 0.003 \text{ in.}}{2}$$

$$A = 35^\circ - \frac{120^\circ}{T}$$

$$J = 0.8D \sin \left(35^\circ - \frac{120^\circ}{T}\right)$$

$$K = 0.8D \cos \left(35^\circ - \frac{120^\circ}{T}\right)$$

$$E = 0.8D + C$$

$$G = 1.24D$$

$$F = 0.8D \cos \left(18^\circ - \frac{56^\circ}{T}\right) + 1.24D \cos \left(17^\circ - \frac{64^\circ}{T}\right) - E$$

$$H = \sqrt{\left[F^2 - \left(G - \frac{P}{2}\right)^2\right]} \quad \text{When } \frac{P}{2} > G, \text{ then } H = F$$

$$O.D. = \text{Outside Diameter} = P \cot \frac{180^\circ}{T} + 2H$$

$$B = 18^\circ - \frac{56^\circ}{T} = \text{Working Arc}$$

Arcs F and E are connected by a common tangent to complete the tooth form.

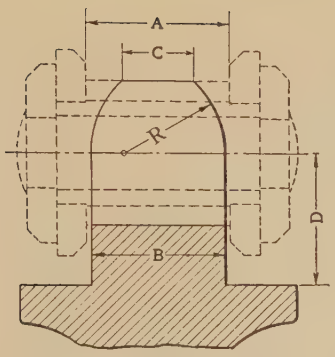


FIG. 3.—SPROCKET DIMENSIONS.

The above equation for the outside diameter of the sprocket blank will cause the teeth to come to a sharp edge. A slightly smaller diameter of blank may be used if desired, as given by the equation

$$O.D. = P\left(0.6 + \cot \frac{180^\circ}{T}\right)$$

In order to insure concentricity of the sprocket and its hub or center, when the two parts are made separate, the sprocket blank should be made an accurate fit over a turned portion of the hub or center, against the flange to which it is bolted. From four to eight bolts are used in securing the front sprocket to its center and a relatively larger number for the rear sprocket.

Chain Pull—If the maximum engine torque is denoted by T, the combined speed reduction ratio of the low gear in the transmission and the bevel gear set by r, the pitch diameter of the sprocket pinion by d and the combined efficiency of the low gear and the bevel gear set by ϵ, then the maximum chain tension is

$$t = \frac{12Tr\epsilon}{100d}$$

Suppose we have a three-ton truck fitted with a four-cylinder $4\frac{1}{2} \times 5$-inch engine which develops a low speed

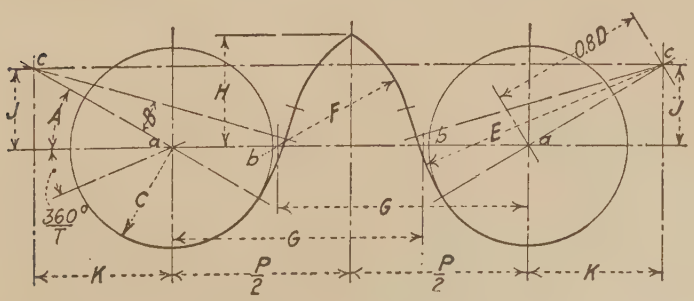


FIG. 4.—SPROCKET TOOTH FORM.

torque of 200 lb.-ft. Suppose the low speed reduction in the change gear is 3.2 and the reduction of the bevel gear set 3. Then, considering the efficiency of the change gear and bevel gear set together to be 90 per cent, the maximum torque on the jackshaft is

$$200 \times 3.2 \times 3 \times 0.90 = 174.5 \text{ lb.-ft.}$$

A suitable size of chain to use is a 1½ in. pitch, 1 in. width of roll. With a 36 in. rear wheel the limiting pitch diameter of the sprocket wheel is about 21 in., hence we could use 45 teeth, which gives a pitch diameter of 21.49 in.; and since the reduction by chain is to be 3 the sprocket pinion must have 15 teeth. This will make the pitch diameter of the sprocket pinion

$$\frac{1.5}{\sin\left(\frac{180^\circ}{15}\right)} = 7.215 \text{ in.}$$

Hence the maximum tension in each chain would be

$$\frac{174.5 \times 12}{7.215} = 2900 \text{ lbs.}$$

Such a chain has an ultimate strength of 19,400 lbs. and therefore would have a factor of safety of 6.7 when transmitting the maximum torque to which the engine could subject it.

Chain-adjusting Rods—Chain-adjusting rods, also known as radius rods, serve a triple purpose. They take up the reaction due to the chain pull, allow of adjusting the slack in the chain, and transmit the driving thrust or braking pull from the rear axle to the frame. These rods must be joined at both ends so as to permit of free play of the springs, and the joint centers should preferably lie in the axes of the sprockets, so that any play of the springs will not affect the sprocket center distances.

Fig. 5 shows a radius rod which has a spherical joint at its forward and a double pivot joint at its rear end. The turnbuckle is provided with a head whose upper and under faces are turned spherically. The upper face of the head bears against the spherical head of a steel button inserted into a drill hole in the wall of the fitting on the jackshaft bearing bracket, and against the under face of the head presses an externally threaded ring screwed into a threaded recess in a boss formed on the fitting, which ring is also pro-

vided with a spherical surface. After adjustment has been made, the nut can be locked in position by means of a clamp screw, and the same locking means is employed for the threaded shank of the turnbuckle.

The joint of the radius rod to the rear axle is of the double pivotal type. A lug is formed on the hub of the brake support which is swiveled on the rear axle, and the rear end of the radius rod is connected to this lug by means of a pin which is held in position either by means of a bolt head and nut or a locking pin, as shown in Fig. 6.

Calculation of Radius Rods—The radius rods act as compression members or columns, and their dimensions should be calculated accordingly. The maximum chain tension can be calculated by the method already explained. Besides this, the rods must transmit the propelling thrust from the rear

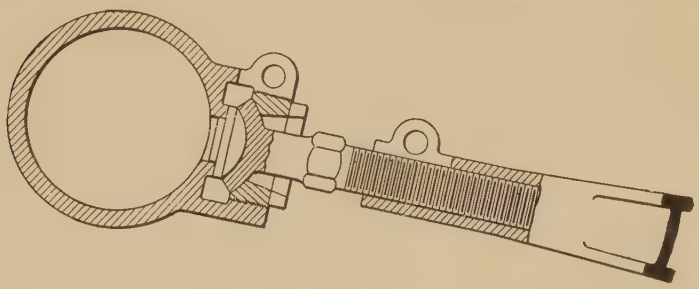


FIG. 5.—RADIUS ROD, FRONT END.

axle to the frame. This propelling thrust can be figured on the basis of 15 per cent of the weight of the vehicle on each of the two rods, for extreme cases. But since the rod makes an angle with the frame, the thrust in the direction of the rod is greater than the propelling thrust, in the ratio of unity to the cosine of this angle.

Thus in the case of the 3-ton truck we found that the maximum chain tension was 2900 lbs. The maximum propelling thrust on each side is about 975 lbs. Assuming that in the full-load position of the springs the radius rods make an angle of 20° with the frame, the thrust along them will be

$$975/0.94 = 1040 \text{ lbs.}$$

and the total pressure on each of the radius rods,

$$2900 + 1040 = 3940 \text{ lbs.}$$

The necessary section can be determined by means of the equation

$$S = \frac{P}{A}\left(1 + \frac{4}{25{,}000}\frac{l^2}{r^2}\right)$$

(Rankine's equation for steel columns free at both ends). In this equation S is the unit compression stress; P, the total pressure on the column; A, the cross sectional area; r, the least radius of gyration of the section, and l the length of the rod or column. In order to use this formula it is necessary to assume a section and determine the value of the stress S, and if this figures out either too high or too low, to make a new assumption.

Effect of Spring Play on Chain Drive—In Fig. 7 is shown a chain drive in diagram in two different positions, the springs

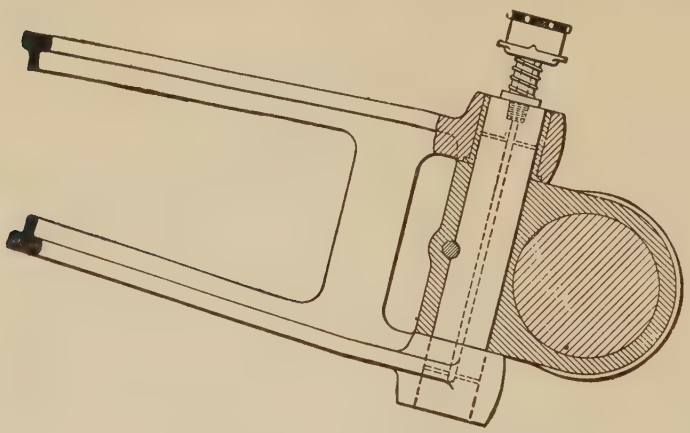


FIG. 6.—RADIUS ROD, REAR END.

being assumed to be distended and compressed, respectively. The sprocket pinion is supposed to have fifteen teeth, and the sprocket wheel forty-five. The distance between centers is assumed to be 28 in., and the total vertical motion of the springs 6 in. We will assume that when the springs are compressed the line of centers is horizontal. Then when the springs rebound, the line of centers moves through an angle α determined by the relation.

$$\sin \alpha = \frac{6}{28}$$

Referring to Fig. 7, a portion of the chain whose length is

$$AB = \frac{\pi D\alpha}{360} \text{ in.}$$

winds up on the sprocket wheel and a portion whose length is

$$CD = \frac{\pi d\alpha}{360} \text{ in.}$$

unwinds from the sprocket pinion. The length of chain between the extreme points of contact on the two sprockets ($E\ B,\ C\ A$) remains the same. Since the length of chain which unwinds from the pinion is not equal to that which

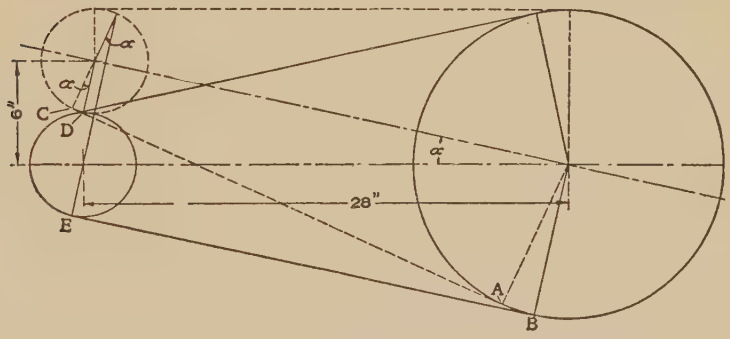


Fig. 7.—Diagram Illustrating Effect of Spring Action on Chain Drive.

winds up on the wheel, if we assume that the wheel remains stationary the sprocket must turn to unwind a length of chain

$$AB - CD = \frac{\pi(D - d)\alpha}{360} \text{ in.}$$

It must turn in the forward direction when the springs rebound and in the backward direction when they compress.

A motion of

$$\frac{\pi(D - d)\alpha}{360} \text{ in.}$$

on the circumference of the sprocket pinion corresponds to an angular motion

$$\Theta = \frac{D - d}{d}\ \alpha \text{ degrees}$$

It will be seen from this that when $D = d$—that is, when the two sprockets are of the same size—the spring action has absolutely no effect on the drive, and the effect is the less the smaller the difference in the sizes of the two sprockets. In the case of our example, since D is substantially equal to 18 in. and d to 6 in., and $a = 12° 21'$, the angular motion of the sprocket pinion corresponding to a spring play of 6 in. is

$$\left(\frac{18 - 6}{6}\right) 12° 21' = 24° 42'$$

The above analysis brings out another reason for making the reduction ratio in the chain drive as small as possible.

CHAPTER XXII

Some Problems of Tractor Design

Farm tractors are used chiefly for plowing, and in their design the requirements in this service must be kept constantly in view. Tractors generally are rated as one-plow, two-plow, three-plow, etc. This rating is based on the number of 14-in. plow bottoms which the tractor is supposed to be capable of hauling. However, the draft required by a 14-in. plow bottom varies between wide limits, depending upon the nature and condition of the soil and upon the depth of plowing. In the Corn Belt of the Middle West the resistance to plowing varies between about 5.5 and 7 lbs. per sq. in., hence with a 14-in. bottom plowing to a depth of 6 in. a draft of from 450 to 600 lb. is required per furrow. In other soils, however, the draft required varies up to 20 lbs. per sq. in. The following table, taken from a catalog of Gray Tractor Co., Inc., gives the average drafts in pounds for a 14-in. plow for various soils and various depths of plowing.

Depth	6 in.	7 in.	8 in.
Sandy	250	300	340
Clover Sod	590	690	785
Clay	675	785	900
* Prairie Sod	1260	1470	1680
Gumbo	1680	1960	2240

Considerable reliable data on the performance of American farm tractors is available by reason of the fact that the State of Nebraska requires that all tractors offered for sale in that state must first be submitted for an official test to the Agricultural Engineering Department of the University of Nebraska. Nearly all tractors on the American market have been tested by that Department, and the results of all tests are published.

If we take the drawbar pulls developed by tractors at plowing speeds in these official tests and divide them by the

* Prairie sod is seldom plowed as deep as 6 in. and the drawbar pull required is therefore seldom as much as here given.

numbers of plows at which the tractors are rated by their manufacturers, we arrive at an average of almost exactly 650 lb. per rated plow.

Number of Gear Speeds—In the earlier tractors the number of forward gear speeds was generally limited to two, but most of the newer tractors are provided with transmissions that afford three forward speeds. If only two forward speeds are available, the high speed is the plowing speed and the low speed is available for emergencies. These two speeds average approximately 2¼ and 3½ m.p.h. At one time a movement was on foot to increase plowing speeds beyond the latter figure, but it seems that the increased power required at the higher speeds, together with the greater wear and tear on the machine nullified the saving on the cost of man-labor.

Where three forward speeds are provided, the intermediate speed, which is generally in the neighborhood of 3 m.p.h., is the plowing speed. The low speed, of about 2¼ m.p.h., is for emergencies, while the high speed, of 4½ m.p.h. or more, is for road work or for farm work other than plowing. The drawbar pull of which the tractor is capable is greatest on the low-speed gear.

Drawbar Horse Power—Expressing the capacity of a tractor in "plows" is quite unsatisfactory, as the "plow" is not a definite unit of power. Therefore, where precision is valued, tractors are rated in drawbar horse power. If the draft which the tractor is capable of delivering at a speed S (in miles per hour) is T lb., then the drawbar horse power is

$$\frac{S \times T \times 5280}{60 \times 33{,}000} = \frac{S \times T}{375}$$

Figuring on a plowing speed of 3.5 m.p.h. and a draft of 650 lb. per plow-bottom, the drawbar horse power is a trifle over 6 per bottom. Similarly, at other plowing speeds with the same draft per plow the drawbar horse powers per plow are as follows:

Plowing speed (m.p.h.)	3	2.5	2.25	2	1.75
Drawbar hp. per plow *	5.2	4.33	3.9	3.46	3

The horse power of the engine with which the tractor is to be equipped must, of course, be considerably greater than the drawbar horse power desired. In the first place, some of the engine power is lost in overcoming the mechanical friction of the transmission, and, in the second, an appreciable amount of power is consumed in moving the tractor itself

* (Based on 650 lbs. draft.)

over the field. The power required to move the tractor itself varies with the condition of the soil, with the size of the wheels and the type of traction device employed. In road traction the diameter of the wheel is not a factor of any consequence as far as the resistance to motion is concerned, but in field work it is. A smaller wheel will sink deeper into the soil and compress the soil more, and this means more power lost in moving the tractor.

Weight and Drawbar Pull—Modern wheeled farm tractors weight about 220 lbs. per maximum drawbar horse power and have a maximum efficiency of about 75 per cent. That is, the maximum horse power which the tractor is capable of delivering on the drawbar is about 75 per cent of the maximum power which its engine will deliver on the dynamometer or on the belt. At a speed of 2¼ m.p.h. such a tractor can exert a maximum drawbar pull equal to about 75 per cent of its weight. However, the drawbar rating given a tractor should be only about two-thirds of its maximum output.

The maximum drawbar pull which a tractor is capable of exerting is evidently limited by three things. First there is the engine power together with the gear reduction. If T is the engine torque in lb.-ft.; r, the total reduction ratio between engine and driving wheels, and R, the driving wheel radius in in., then, allowing 16 per cent for transmission losses, the torque on the drivers is

$$T_d = 0.84 \times T \times r \text{ lb.-ft.},$$

and the tangential force or total traction of the drivers,

$$F = 0.84 \times T \times r \times 12/R = 10Tr/R \text{ (appr.)}$$

The next limit to the maximum drawbar pull is the total weight of the tractor, for this limits the adherence of the wheels to the ground. This adherence to the ground is not a very definite magnitude, for it varies with the nature of the soil and with the type of traction device used. The third factor which limits the drawbar pull is the distribution of the weight between the front and rear axles and the spacing of these axles, or the wheelbase.

"Rearing" Tendency of Tractors—When a tractive effort is exerted by the driving wheels, there is a reaction on the tractor frame which tends to cause it to rise in front, so that virtually weight is removed from the front wheels and added to that already on the rear wheels. This shifting of weight is proportional to the traction force or the circumferential force on the rear wheels, and to the radius of the

wheels, and it is inversely proportional to the wheelbase. It is permissible and desirable to so distribute the weight between front and rear wheels, and use such a driving wheel diameter and wheelbase, that when the tractor exerts its maximum traction, all of the weight is removed from the front wheels (considering only the effect of the torque reaction). Since the drawbar is connected to the tractor at a point below the driving wheel axis, the drawbar pull produces a moment which keeps the front of the tractor down and keeps the machine from turning over backward. However, turning over backward is possible when the tractor is not pulling any load, as when the machine is stuck in a ditch, etc., and the driver puts on full power to get it out. This can be prevented only by providing the machine with a substantial platform back of the axle which hits the ground after the front wheels have left the ground a little ways.

The problem now is to determine the proper relations between the weights on the front and the rear axle, the driving wheel radius and the wheelbase. With a transmission efficiency of 84 per cent, and a drawbar efficiency of 75 per cent, 1.12 hp. will be delivered at the rims of the drivers for each drawbar hp. and a total weight of 220 lb. If the low speed or emergency speed is 2¼ m.p.h., which corresponds to 198 ft. p.m. the maximum traction per drawbar horse power will be

$$\frac{1.12 \times 33{,}000}{198} = 186 \text{ lbs.}$$

This tangential force on the driving wheels will remove from the front wheels a weight 186 R/W where R is the driving wheel radius and W the wheelbase, both in in. This, as stated above, must not exceed the weight originally on the front wheels. Dividing this by the total weight of 220 lbs. gives us the minimum proportion which the weight on the front wheels must bear to the total weight.

$$\text{Proportion of weight on front wheels} = (186/220)\ (R/W)$$

This gives the following values for the minimum proportional weights W_f on front wheels for different ratios of driving wheel radius to length of wheelbase:

$R/W =$	0.30	0.32	0.34	0.36	0.38	0.40
$W_f =$	25.3	27.1	28.7	30.5	32.1	33.8

At a speed of 2¼ m.p.h. one drawbar horse power corresponds to a drawbar pull of 166.5 lbs., which is a little

over 75 per cent of the tractor weight per drawbar hp. At a plowing speed of 3½ m.p.h. the drawbar pull corresponding to one horse power is 107 lbs., or a little over 48 per cent of the tractor weight per drawbar hp.

General Arrangement of Parts—The engine of the tractor may be arranged in two general ways, either coaxial with the tractor, or transversely across it. The first arrangement is the most popular, and the chief reason for using the transverse arrangement of the engine appears to be that that permits of doing away with bevel gears in the transmission. If the engine is mounted crosswise in the tractor the transmission can be entirely by spur gears. Bevel gears, of course, are not absolutely essential with an engine in the normal fore-and-aft position, as the change of direction in the power flow can be obtained also by means of worm gearing.

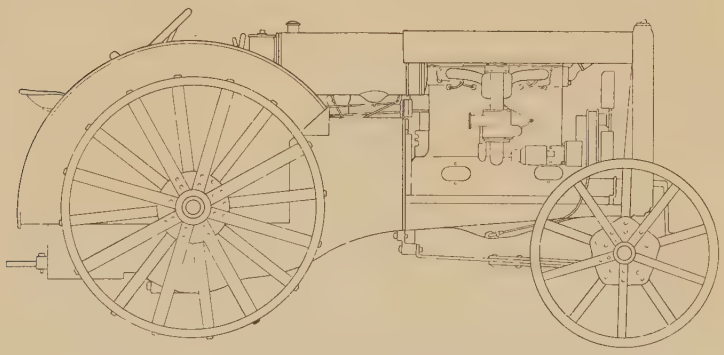


FIG. 1.—SIDE VIEW OF TYPICAL WHEELED TRACTOR.

A disadvantage of the crosswise arrangement is that it brings the center of gravity of the engine closer to the rear axle and therefore lessens the proportion of the weight on the front axle, if the transmission is by toothed gearing throughout. This can be avoided by using a first reduction by chain, which permits of placing the engine farther forward. In one or two tractors on the market the final reduction is by chains.

A side elevation of a tractor of the predominant type, with the engine located centrally at the front (with its crankshaft extending fore-and-aft) is shown in Fig. 1.

Wheeled tractors are divided into tractors with frames (generally made of rolled steel sections) and frameless tractors, in which the engine crankcase and the housing of the

transmission and final drive gears take the place of the frame. The greatest weight efficiency is obtained when the housings perform the functions of a frame. One objection to this type is that dismounting the engine for repairs ("splitting the tractor") is a task that is quite difficult to carry out on the farm. For this reason a development of the original frameless type is sometimes used, in which the transmission housing or rear axle center housing is provided with a forwardly extending frame of either pressed steel, rolled steel or cast iron, on which the engine (and possibly also the gearbox) is set. The engine then can be removed from the tractor by means of a hoist without letting the frame structure down. A cross-section of a trough shaped cast frame is shown in Fig. 1-A.

Clutches—The types of clutch most extensively used on farm tractors are the single plate and the multiple disc. The principles underlying the design of tractor clutches are the same as those on which the design of clutches for motor vehicles is based. In plowing, of course, gears are not shifted while the tractor is in motion, and there is therefore not the same need for reducing the inertia of the clutch driven member as in a motor-vehicle clutch.

However, there is a pronounced tendency to make farm tractors available also for road traction and general industrial purposes, and in a number of instances such tractors have been fitted with pneumatic tires for road work and have been equipped with four-speed transmissions. When traveling on the road at 8–10 m.p.h., it is, of course, necessary to change gears while in motion, and the clutch requirements are then substantially the same as in regular road vehicles. This may explain why the multiple disc clutch, which was quite prevalent in tractor work in the early twenties, is giving way more and more to the single-disc type.

One type of disc clutch widely used on tractors, and which is controlled by a hand-lever, was described in the Chapter on Friction Clutches.

Multiple-disc-type clutches are used to a certain extent, but their design is very much the same as that of similar clutches for road vehicles. In one particular case the clutch for a four-cylinder 4½ by 6 in. engine comprises six driving and five driven discs of 8 in. inside and 10½ in. outside diameter. This clutch has to transmit the power of an engine developing a maximum torque of about 160 lb.-ft. and a torque of 142 lb.-ft. at its normal speed of 1000 r.p.m.

An expanding-shoe-type of clutch is also used to a certain extent. It has two shoes, pivoted on studs parallel with the

clutch axis that are fastened into the clutch spider. In order to prevent the action of the shoes being influenced by centrifugal force, they are provided with counter-weights located on the opposite sides of the studs from the shoes. The shoes are pressed into contact with the clutch drum by means of a

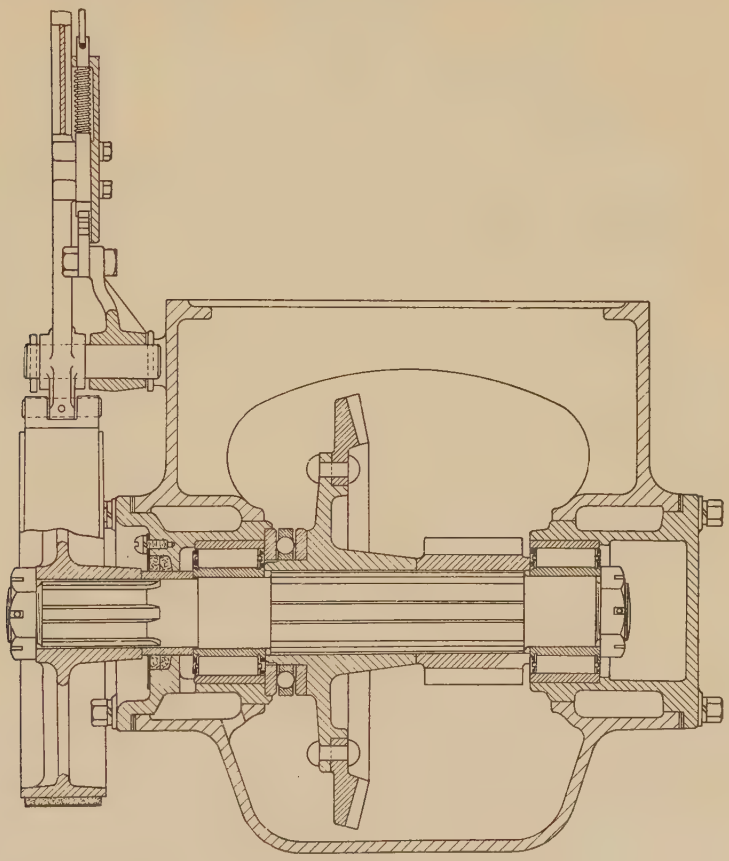


FIG. 1-A.—CROSS SECTION THROUGH TROUGH-SHAPED FRAME (McCormick-Deering).

toggle mechanism, and are also positively withdrawn from the drum by this same mechanism.

Gear Reduction Ratios Required—The engines of farm tractors, as a rule, run at between 800 and 1200 r.p.m., although there are a few examples of both lower and higher

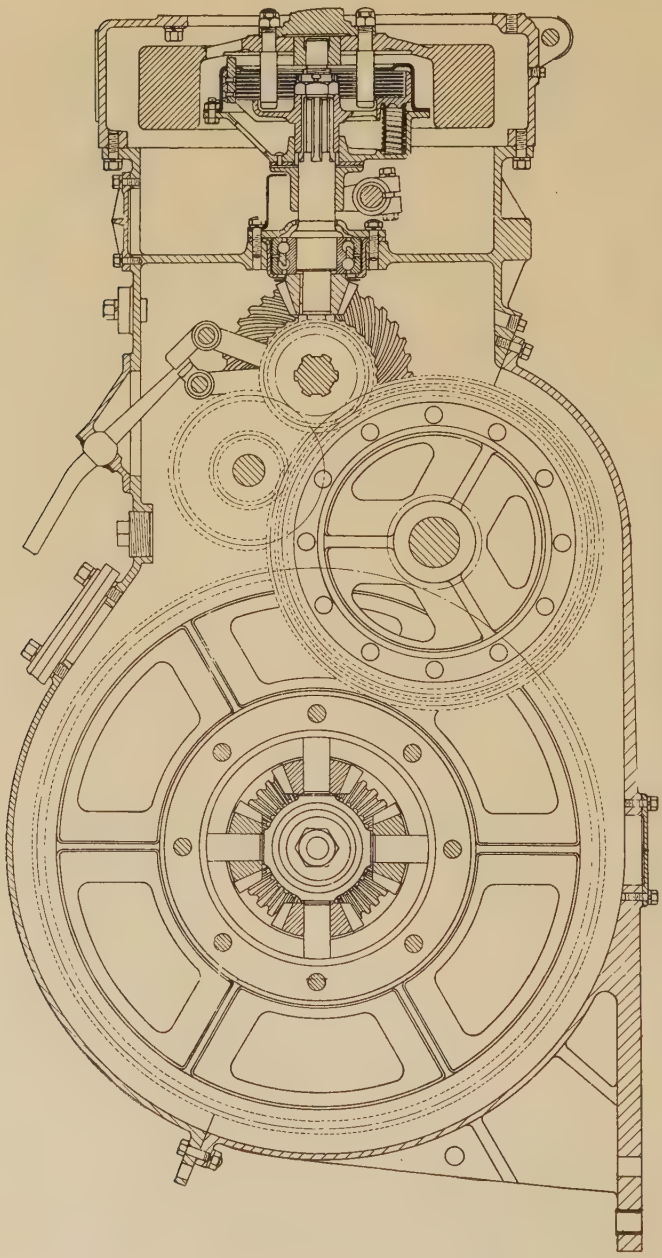


FIG. 2.—TWO-SPEED-AND-REVERSE TRACTOR TRANSMISSION IN VERTICAL SECTION.

engine speeds. It is not likely that the speeds of these engines will be increased materially in the future, as there is little to be gained. An increase in engine speed would permit of an increase in engine power without an increase in weight, which could be translated either into higher tractor speed or greater drawbar pull. But the speed of the tractor is limited by the services in which it is used, and to really obtain increased drawbar pull it is necessary to have greater tractor weight. On the other hand, a moderate engine speed has the advantage of reducing the wear and tear on the engine.

The reduction ratio of the transmission depends upon three factors, namely, the engine speed, the diameter of the driving wheels, and the tractor speed. The reduction ratios for low speed in many cases are very large. Take, for example, a tractor with 50-in. driving wheels and an engine turning over at 1000 r.p.m. At 2¼ m.p.h., allowing 5 per cent for slippage, such a tractor must have a gear reduction of 63 to 1. A number of tractors, however, have considerably lower low speeds, and the Fordson tractor, for instance, has a low speed reduction ratio of 82 to 1, even though its drivers are of comparatively small diameter.

Arrangement of Drive—An arrangement of the transmission that has been quite popular for tractors with two forward speeds is as follows (Figs. 2 and 3): On the clutch shaft is mounted a bevel pinion (usually of the spiral bevel type, on account of the high speed at which it runs), which meshes with a corresponding gear on a transverse shaft. This transverse shaft is splined and carries two sliding pinions adapted to be meshed with gears on a parallel shaft farther back and somewhat lower down. The second transverse shaft carries, in addition to the two gears mentioned, a spur pinion meshing with a large gear on the differential in the rear axle. Preferably the reduction ratio is increased for each successive pair of gears. In the final drive gears a reduction ratio of 6 to 1 may be obtained, and in the bevel gears of about 2.5 to 1, which makes it necessary, if a total reduction of about 75 to 1 is required for the emergency and 48 to 1 for the plowing speed, to use sliding gear sets having ratios of 5 to 1 and 3.2 to 1.

Using bevel gears for the first reduction rather than for a later one, has several advantages. Bevel gears generally overhang their bearings, and therefore give rise to greater bearing loads than spur gears, besides which they create end thrust. By placing the bevel gears at the beginning of the train, where the torque is still small, the increase in radial bearing loads and the thrust load will both be minimized.

Moreover, the same bevel pinion from which the power is transmitted to the driving wheels, can be more advantageously used for driving the belt-pulley shaft, which should preferably turn at about crankshaft speed. In order to keep down the horizontal distance between the first transverse shaft (the splined shaft) and the rear axle axis—and thus the length of the tractor—the several transverse gear shafts are placed at different levels.

Figure 4 shows the transmission gearing of a tractor having bull gears secured to the rear driving wheels. This was the standard method of final drive in the earlier tractors, but is becoming less and less common, as it is difficult to enclose gears at the driving wheels effectively. In the earlier tractors the bull gears were usually completely exposed, the gears being made of cast steel, with the teeth cast on and of comparatively large dimensions, so that the tooth pressures would not be excessive in view of the unhardened tooth surfaces. In the transmission here represented the bull gears are cut with internal teeth and are fully enclosed.

Three-Speed Transmission—An arrangement of the transmission somewhat similar to that represented by Figs. 2 and 3 but designed to give three forward changes of speed, is shown in Fig. 5 in plan view. In this case, however, a quadruple reduction to the rear axle is employed. With four steps in the transmission one naturally expects a somewhat greater power loss, but the arrangement has several advantages. It permits of a small first reduction, so that the splined shaft runs at near crankshaft speed, and the belt pulley is mounted on an extension of this shaft. Secondly, each individual reduction ratio will be smaller, and it is not necessary to use such a fine pitch, nor so small a number of teeth, in the pinions. A coarser pitch gives greater tooth strength, and a larger number of teeth in the pinion gives better tooth action. Unfortunately, with three sliding pinions on the splined shaft in addition to the bevel gear and spaces between the pinions to permit of sliding them along the shaft for meshing and unmeshing, the necessary width of the transmission housing becomes rather large, and since the belt pulley must be located between it and one of the driving wheels, the parts become rather cramped in the transverse direction. For this reason some engineers prefer to have the splined shaft extend fore and aft in line with the engine crankshaft, and a secondary shaft below it, from which the power is transmitted to the first transverse shaft by bevel gears.

Sliding Pinions on Longitudinal Shaft—This arrangement is used in the McCormick-Deering tractor of the Inter-

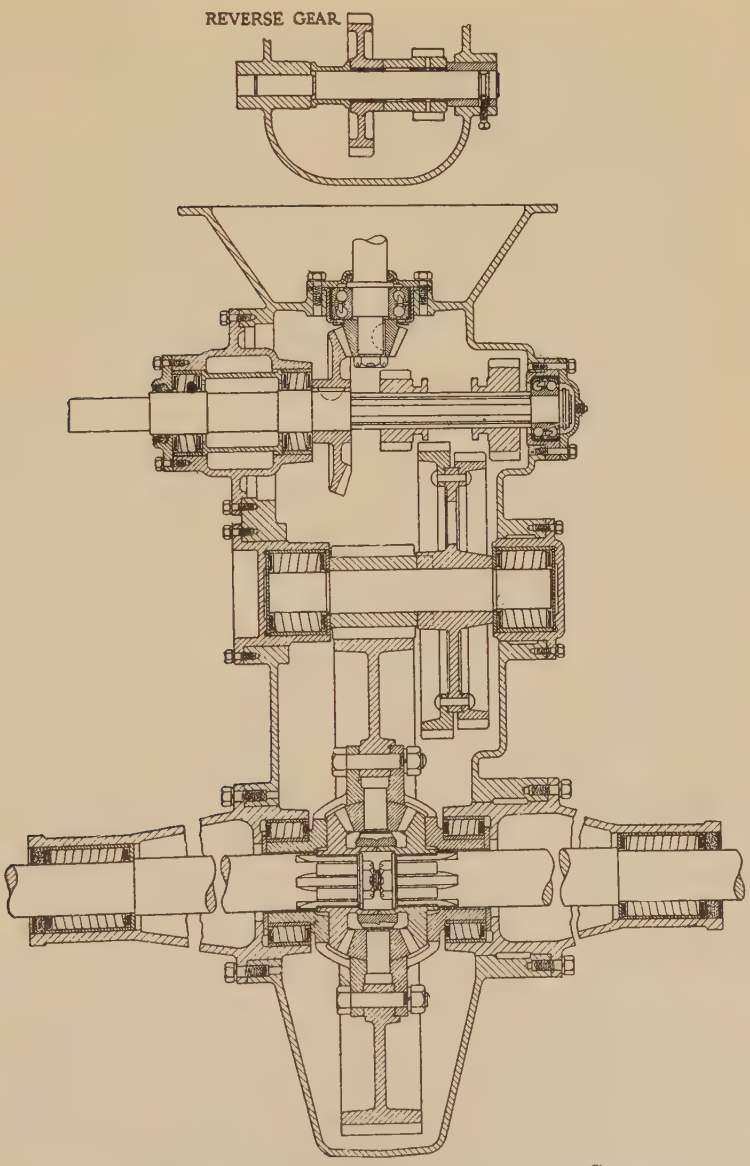


FIG. 3.—HORIZONTAL SECTION OF TRANSMISSION SHOWN IN VERTICAL SECTION IN FIG. 2.

national Harvester Co., of which a sectional view is shown in Fig. 6. In this case there are three parallel shafts, one above the other. The power comes in from the friction clutch at the right hand end of the splined intermediate shaft, the friction clutch shaft connecting to this shaft through a gear-type of positive clutch. On this shaft there are three sliding pinions, which are adapted to mesh with three corresponding gears on the lower shaft and with a reverse idler which is indicated in dotted lines. The sliding pinion on the right is shown in the drawing in mesh with a gear on the uppermost shaft, from which latter the belt-pulley shaft is driven through a pair of bevel gears. If this pinion is shifted farther to

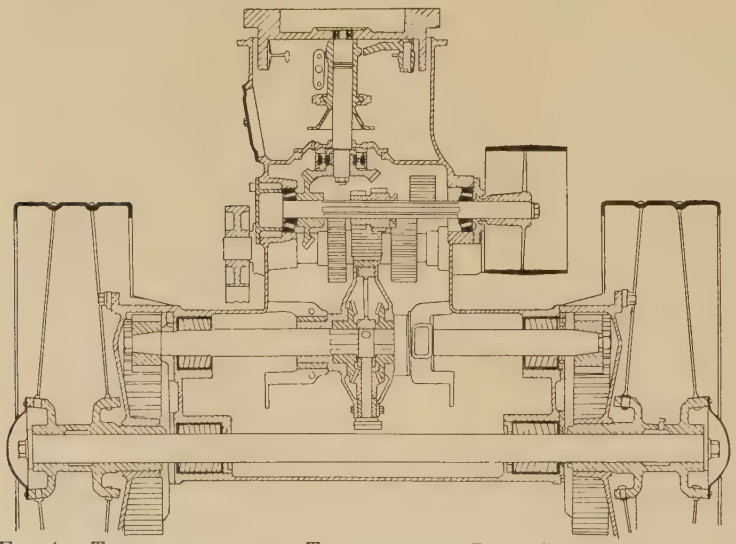


Fig. 4.—Transmission of a Tractor with Bull Gears on Driving Wheels.

the right it will mesh with the reverse idler to give the reverse speed, while if it is moved to the left it will mesh with the high-speed gear on the lower shaft. The other two sliding pinions form a single unit; when slid to the right they give the intermediate speed, while when slid to the left they give the low forward speed. The narrow-faced gear on the lower shaft at the right is merely an oil-thrower. The gear on the upper shaft through which the belt pulley is driven is loose on its shaft and can be secured thereon by means of a jaw clutch operated by a lever on top of the gear box. Thus the

belt pulley will be stationary when the tractor is delivering power on the drawbar, which is desirable.

All three of the shafts in the transmission housing are supported on cylindrical roller bearings, while the end thrust due to the bevel gears is taken up on ball thrust bearings. The transmission housing in this case does not form a part of the tractor frame but is provided with feet by which it is supported on a cast trough shaped frame that forms the rear axle center housing and extends forward all the way to the

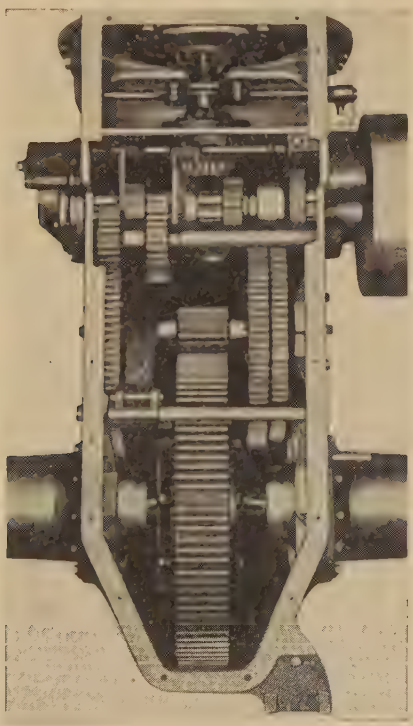


FIG. 5.—THREE-SPEED TRACTOR TRANSMISSION IN PLAN VIEW.

front axle. The two slider bars extend through the rear wall of the housing and are operated by a lever with ball and socket mounting.

Fordson Transmission—In the Fordson tractor the transmission is also arranged with its shafts extending fore and aft, but in this case the gears remain constantly in mesh and the different speeds are obtained by locking one member of a

pair of gears to its shaft. Moreover, the arrangement of gears is such that for the two higher forward speeds, power is transmitted through only a single pair of gears in the transmission, while for the low forward (emergency) speed and the reverse motion the power is transmitted through three pairs of gears.

Referring to Fig. 7, which is a vertical section through this transmission, the upper shaft receives power from the engine at the left, while the lower shaft has an extension on

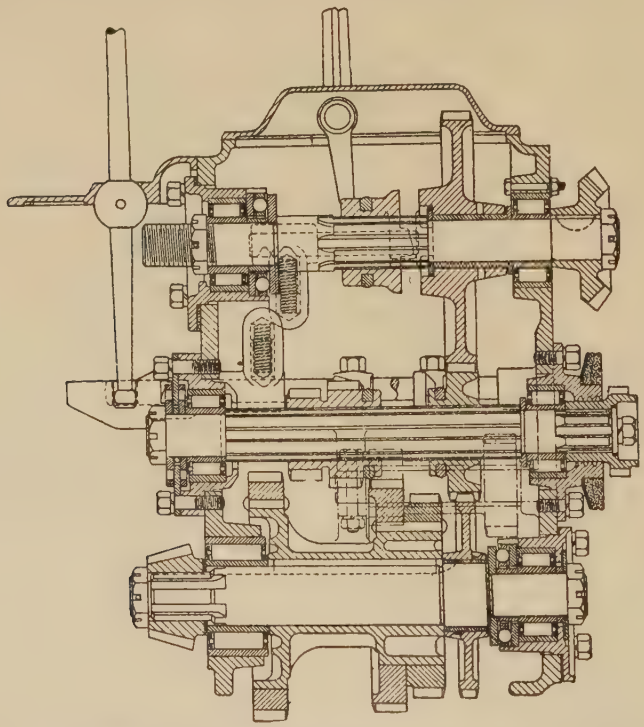


Fig. 6.—Three-Speed Tractor Transmission with Longitudinal Shafts.

the right which carries the worm meshing with the rear-axle worm wheel. The pinion at the upper left and the gear at the lower right are loose on the shafts in the gearbox, and may be secured thereto by means of gear-type positive clutches, but they have their own shafts which are mounted in bearings in the wall of the housing and extend outside the housing.

For the low forward speed the power is transmitted from the transmission drive pinion at the upper left to its mating gear on the shaft below; from the lower shaft to the upper through the pair of gears second from the right, and then to the transmission drive shaft through the pair of gears at the extreme right.

For the second forward speed (the regular plowing speed) the sliding member on the upper shaft is shifted toward the left, locking the pinion at the left of the upper shaft to this shaft. The power is then transmitted from the upper shaft through the pair of gears at the extreme right. For the high speed forward the sliding member on the lower shaft is shifted to the right, and the power is then transmitted through the pair of gears at the left with only a very slight reduction in speed (24/23).

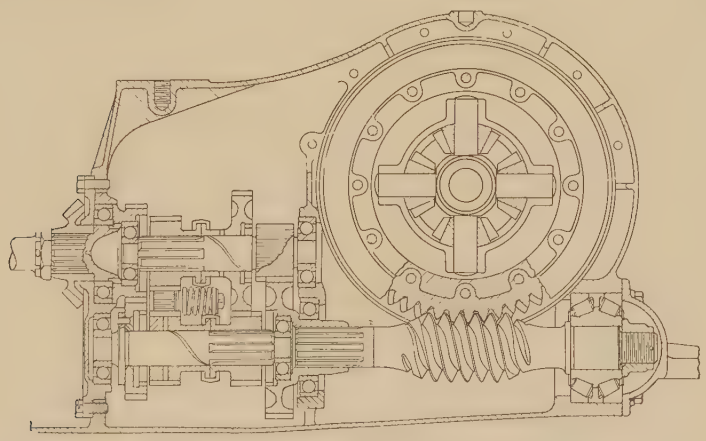


FIG. 7.—TRANSMISSION OF FORDSON TRACTOR.

To obtain the reverse, the sliding member on the lower shaft is shifted toward the left to lock the gear at the left to the lower shaft. Power is then transmitted first through the pair of gears at the left, then through a train of three gears adjacent to these, and finally through the pair of gears at the right.

Dimensions of Gears—It was shown in the Chapter on Sliding-Gear Transmissions that the diametral pitch for use in sliding gears should vary inversely as the cube root of the maximum torque on the pinion. A suitable pitch for the gears of tractor transmissions is obtained from the equation

$$p = 30/\sqrt[3]{T},$$

where T is the maximum torque in lb.-ft. on the pinion shaft or splined shaft of the transmission. This equation gives the following ranges of pinion-shaft torque for different diametral pitches:

Diametral pitch.....	6	5	4½	4
Torque range (lb.-ft.)	100-162	162-252	252-350	350-512

Similarly the diametral pitch for the bull gears or final drive gears can be determined from the equation

$$p = 40/\sqrt[3]{T},$$

where T is the maximum torque in plowing gear on each of the bull pinions. This gives the following torque ranges for the different diametral pitches:

Diametral pitch	5	4½	4	3½	3
Torque, lb.-ft.	400-600	600-833	833-1210	1210-1870	1870-3080

The gears should be calculated for resistance to wear by means of the equation developed in the Chapter on Some Problems of Toothed Gearing, which for 20°-pressure angle teeth takes the form

$$S_c = 3900\sqrt{\frac{P(1+\rho)}{fR}}$$

In this equation the compression stress S_c should not exceed 100,000 lbs. per sq. in. in pinions running at engine speed or near that speed and which have to carry load continuously; 150,000 lbs. per sq. in. for pinions running at engine speed and which are intended for emergency service only; 150,000 lbs. per sq. in. for the final reduction gears when the maximum engine power is being transmitted through the regular plowing gear, and 200,000 lbs. per sq. in. when the maximum engine power is being transmitted through the emergency gear. These figures apply to cut steel gears either oil-hardened or case-hardened.

Since there are normally three or four gear reductions in tractors with live axles, it is possible to place the differential gear either in the rear axle or on the last intermediate shaft. Since the bull gears give a reduction of the order of 6 to 1, if the differential is on the last intermediate shaft it has to handle only about one-sixth the torque, and therefore can be

made much lighter. Where this arrangement is used it is necessary, however, to have two pairs of bull gears, each of the bull pinions being carried by one of the differential side gears.

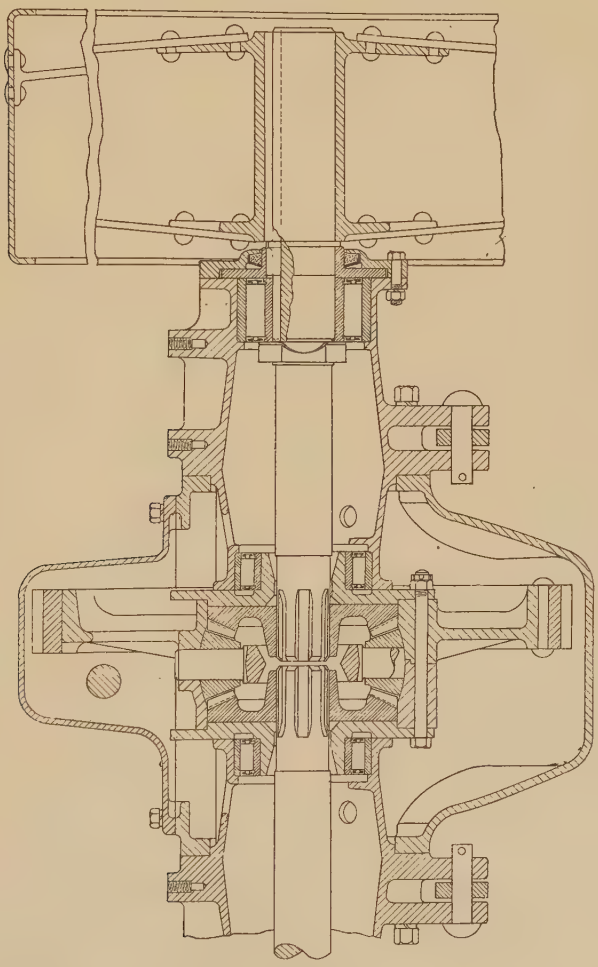


FIG. 8.—REAR AXLE OF MCCORMICK-DEERING TRACTOR.

Rear Axles—The final drive of farm tractors, as a rule, is effected by gears mounted either in a central case or else mounted on the wheels and protected by a suitable housing. In the first case a live axle is used, which is generally of the semi-floating type, while in the latter case the rear axle

is a plain carrying axle. The central housing enclosing the bull gears generally forms a part of the frame. It is provided with large openings in the sides through which the axle shafts pass and against which cast axle housings of considerably larger diameter than the shafts are bolted. A sectional view

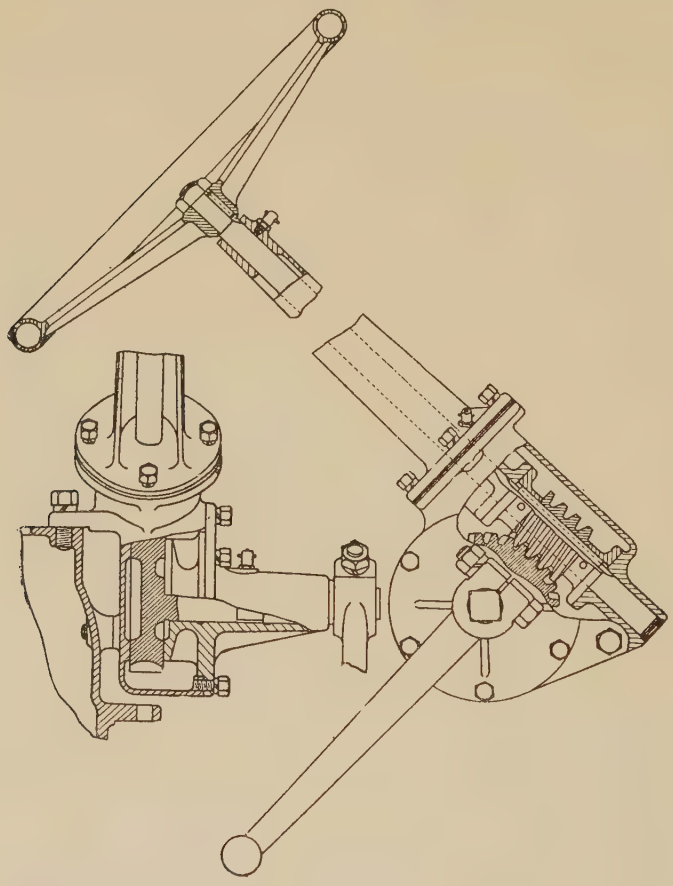


Fig. 9.—Steering Gear of McCormick-Deering Tractor.

of such an axle is shown in Fig. 8. The axle housings project into the case and form the mounts for the differential bearings, while another set of bearings are set into the axle housings at their outer ends. In this particular design a form of cylindrical roller bearing is used, and end thrust is taken

up on a hardened steel thrust ring mounted on the axle shaft and located between the end of the tube and a bolted-on cap.

Front Axles—In order that the tractor may accommodate itself to uneven ground without unduly stressing its members, the front axle is usually pivoted to the frame or engine at its middle, so it can swing freely in a vertical plane. This necessitates tieing the ends of the axle to a point some distance back—usually the engine flywheel housing—by means of radius rods. Otherwise the axles are designed generally

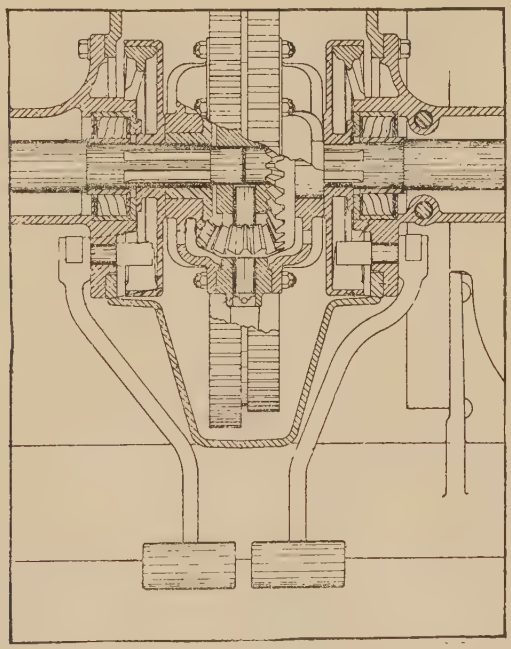


Fig. 10.—Brakes on Differential to Help in Steering.

along the same lines as truck axles. Steering gears also are of the same type as used on motor vehicles. The steering gear is generally mounted centrally on the frame, but if mounted off center it is located on the right side, so as to give the operator a better view of the furrow. A typical tractor steering gear is shown in Fig. 9.

Since the weight on the front wheels when the tractor is pulling hard is comparatively small, the steering is not very positive, and in a few tractors, in order to assist in steering,

brakes are placed on the side gears of the differential, which latter are provided with brake drums for the purpose. Each brake can be independently operated by its own pedal. By applying the brake on one differential side gear, the driving wheel on that side will be retarded and the driving effort of the other wheel then has a natural tendency to cause the tractor to turn around, thus assisting the steering gear. A design of this type is shown in Fig. 10.

Belt Pulleys—The design of belt pulleys is complicated by the fact that power-operated farm machinery is designed for different belt speeds. At the present time three belt speeds are standardized, viz., 2600, 3000 and 3500 ft. p. m., but it is recommended that any new equipment be designed for a belt speed of 3000 ft. p. m. so the prospects are that this will eventually become the sole standard.

From the belt speed, the governed speed of the engine, and the ratio between crankshaft speed and belt-pulley speeds the necessary diameter of the pulley can be readily calculated. However, the pulley must not have a diameter smaller than 12 in., as the sharp bend given the belt by a smaller pulley has a destructive effect on it; and it should be ½ in. wider than the belt to be used. In Fig. 11 is shown the belt pulley with its shaft, drive gear and housing of the McCormick-Deering tractor. The pulley is mounted on the splined end of the shaft, a nut on the end of the shaft forcing its hub and the inner race of the roller bearing against a substantial collar on the shaft. This shaft can be disconnected so it will not turn when the tractor is moving. Where no clutch is provided for the pulley shaft the pulley is sometimes split and clamped to its shaft by bolts, so it can be readily removed when the tractor is to be used for drawbar work.

Side Draft—A rather important point in the design of a tractor is the selection of the point at which the drawbar connects to the tractor frame. For obvious reasons it is desirable that the drawbar load should be transferred to the tractor at some point in the vertical plane through its fore-and-aft axis, and that the load should be in line with this axis. If these conditions are fulfilled there will be no side draft. The drawbar load will not produce any end thrust on the wheels and bearings, and will not tend to cause the tractor to leave its straight-ahead course.

But in many cases it is difficult and even impossible to fully meet these conditions. This applies particularly to tractors of small capacity. Let us take the case of a two-plow tractor. The center of load or draft on a two-bottom

plow is about 18 in. from the edge of the furrow. But the width of the tractor over the rear wheels is about 50 in., and if the right hand driving wheel were run close up to the furrow, the axis of the tractor would be 25 in. from the edge of the furrow and therefore offset 7 in. with respect to the center of draft of the plow.

By moving the point of attachment of the drawbar to the tractor over to the right, the line of pull can be made parallel with the direction of tractor motion. But in that case the resistance to motion of the right driving wheel will be much greater than the resistance to motion of the left wheel. The right driver therefore will slip more than the left one, with the result that the latter tends to run ahead of the former

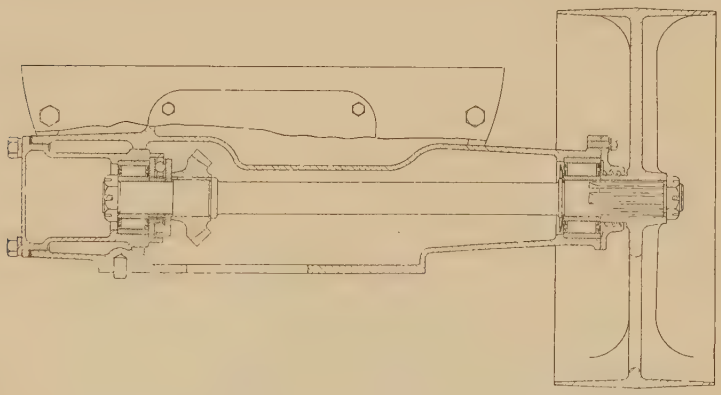


FIG. 11.—BELT-PULLEY DRIVE.

and to cause the tractor to turn around. This effect is plainly noticeable to the operator, who must constantly pull on the steering wheel to hold the tractor from running into the furrow.

Another plan consists in running the tractor with the right wheels in the furrow. This obviates side draft in most cases and may even produce a slight negative side draft, tending to pull the plow toward the furrow, which is said to be desirable in its effect on the plowing. However, running one pair of wheels of the ordinary tractor in the furrow is not good for the tractor, as it tilts it to an appreciable extent, thereby producing thrust loads on the bearings and in some types of tractor interfering with proper engine lubrication. It is, of course, also rather unpleasant for the operator to

ride on a tractor listing over to one side. A number of tractors have been designed so the right wheel could be lowered an amount equal to the depth of the furrow, and the tractor then would be on an even keel when this wheel ran in the furrow, but this feature cannot be readily incorporated in the standard type of tractor with two rear driving wheels.

Point of Hitch—About the only way to reduce the amount of the side draft with tractors whose width over the drivers considerably exceeds the width of the strip of soil turned in one pass consists in lengthening the drawbar and attaching it to a point on the tractor frame comparatively far forward. This plan is now widely followed. The drawbar is connected to the under side of the transmission housing (which usually

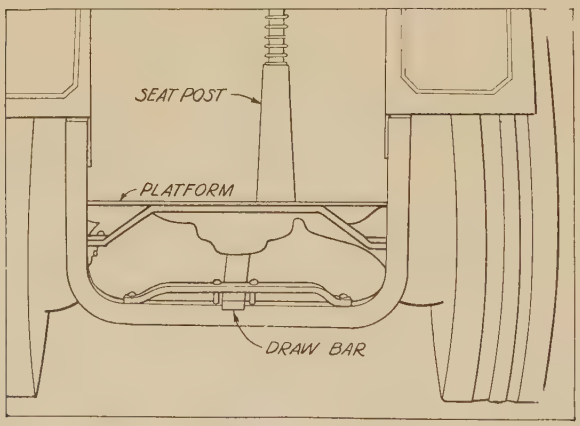


Fig. 12.—Drawbar Adjustable in a Horizontal Plane.

constitutes part of the frame) at a point some distance in front of the rear axle, in such a manner that it can swivel freely in a horizontal plane. To the rear end of the frame or to the operator's platform are secured two transverse guide bars between which the rear end of the drawbar may move freely. However, a number of holes are drilled through these guide bars and bolts can be put through them to hold the rear end of the drawbar in various positions relative to the axis of the tractor. Such an arrangement of the drawbar is shown in Fig. 12.

Adjustments of the hitching point of tractors for plows with different numbers of bottoms have been standardized as follows: The standard height for drawbars that are non-

adjustable vertically is 15 in. above the ground. If the drawbar is adjustable vertically the range of adjustment should be from 13 in. to 18 in. The maximum horizontal distance from the furrow wall to which the rear end of the drawbar can be adjusted (which is often required when the right hand wheel runs on the land) is 30 in. for a two-bottom 14-in. plow; 32 in. for a three-bottom and 40 in. for a four-bottom; the minimum distance of the hitching point from

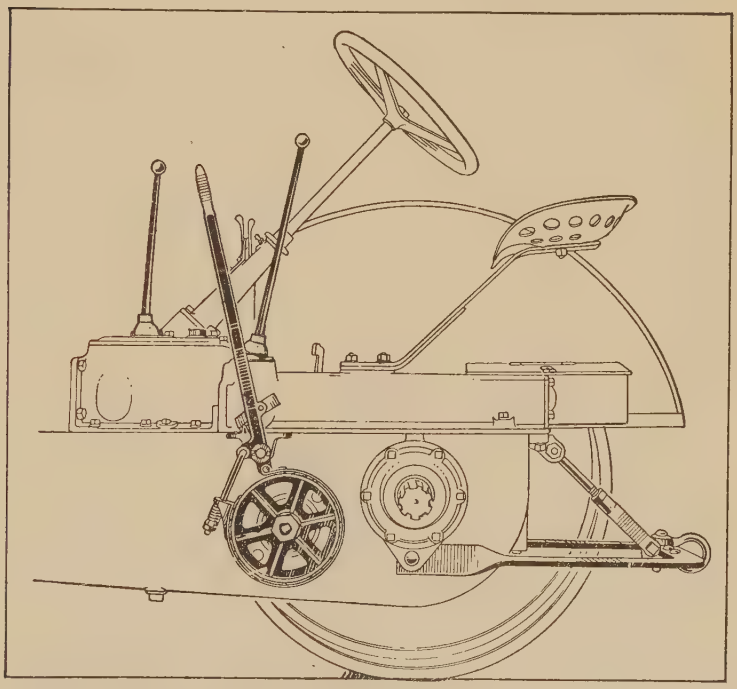


FIG. 13.—HORIZONTALLY AND VERTICALLY ADJUSTABLE DRAWBAR.

the edge of the furrow (used when the tractor operates in the furrow) is 15 in. for a two-bottom, 20 in. for a three-bottom, and 24 in. for a four-bottom plow. The best average distance of the hitching point from the edge of the furrow is 20 in. for a two-plow tractor; 26 in. in the case of a three-plow, and 32 in. in the case of a four-plow.

A drawbar which is adjustable vertically is used on McCormick-Deering tractors and is illustrated in Fig. 13. It is made of strip steel and of the yoke type, the ends of the

yoke being pivoted to lugs cast on the under side of the axle housings. The central part of the yoke is connected to the driver's platform by means of an adjustable link, and its height can be varied by changing the length of this link, which is in two parts that are screwed together. This central part of the drawbar has a number of holes in it, and a clevis to which the implement drawbar is attached can be connected to the drawbar at any of these holes, thus effecting adjustment in a horizontal plane.

In some cases the drawbar is spring-cushioned, in order to relieve the tractor as well as the implement of shock in case of a sudden increase in soil resistance. The drawbar then passes through an eye formed on the under side of the tractor frame and is surrounded by a coiled spring in front of this eye, the forward end of the drawbar being provided with a nut and perhaps a washer which bears against the spring. Excessive increases in the resistance encountered by the implement are guarded against by using a wooden plug for connecting the implement drawbar to the tractor drawbar, which will shear off if the drawbar pull greatly exceeds the maximum which the tractor can deliver steadily; or an automatic drawbar hitch, of which a number have been invented, and which depend upon the compression of a coiled spring to unhitch the implement.

Figure 13 which illustrates the drawbar used on the McCormick-Deering tractor, also shows the arrangement of the controls on this tractor. The steering post with the steering wheel is mounted in the center and has about the same rake as in an automobile. Of the two rearwardly inclined levers the forward one is the gearshift lever and the rear one the belt-control lever, which directly operates the positive clutch, locking the belt-pulley gear to its shaft. The forwardly inclined lever is the brake lever, as plainly shown.

Wheel Sizes—The wheels used on tractors are of the so-called agricultural type, with cast hubs, strip-steel or round steel spokes, and rolled steel rims. Strip steel spokes, if used, are secured to both the rim and flanges on the hubs by means of rivets, while round steel spokes are riveted into the rims and cast or welded in the hubs. In order to increase the rigidity of the wheels the rims are flanged over at the sides, and sometimes, as in the case of the French & Hecht wheel illustrated in Fig. 14, both flanged and grooved. It is desirable to use fairly large wheel diameters and widths of rim, as this will reduce the packing of the soil and the resulting power loss. The wheels are usually so dimensioned

that the weight on them per inch diameter and per inch of width lies between 3 and 3½ lbs. The following table gives approximate average wheel sizes for tractors of different capacities:

Capacity	Front Wheels	Rear Wheels
Two-plow	30 × 5	46 × 10
Three-plow	32 × 6	50 × 12
Four-plow	34 × 6	54 × 15
Five-plow	36 × 8	60 × 18

While the weight distribution between front and rear wheels with the tractor stationary is usually in the proportion of about 1 to 2, the ratio between the projected areas (diameter times width) of the wheels is made greater, from

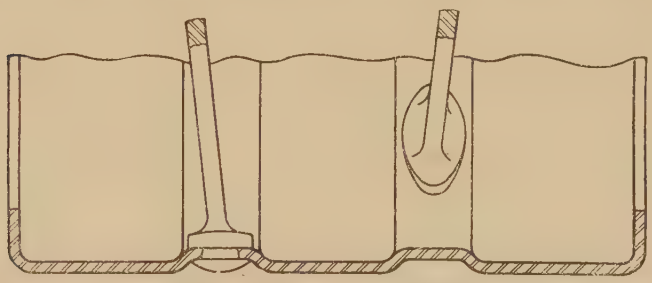


FIG. 14.—GROOVED RIM OF FRENCH & HECHT TRACTOR WHEEL.

1:3 to 1:4, because in operation the front wheels are relieved while the rear wheels are more heavily loaded.

On the early tractors wheels of exceedingly large diameters were used, and the general tendency during the period of tractor development has been toward smaller wheels, even though they pack the soil more and therefore cause more power loss. One advantage of small wheels is that they do not require so large a gear reduction. In the Fordson tractor, for instance, which has drivers 42 in. in diameter, the gear reduction for the plowing speed (44.5) is obtained in two steps, by one pair of spur gears and a worm-wheel set, while most other tractors make use of three reductions, and some even of four. In the Fordson a triple-lead worm is used, in conjunction with a 51-tooth wheel.

Front wheels are usually provided with a skid ring, which digs into the soil and steadies the front end. This ring is

usually turned up from angle iron and is secured to the rim of the wheel by bolts. Under most conditions of plowing it is preferable to have the skid ring at the center of the wheel rim, but for plowing in gumbo it is better to have it near the inner edge, as the wheel will then clear itself better. A typical front wheel with angle-iron skid ring, together with the steering knuckle and axle end, is shown in Fig. 15.

Rear wheels must be provided with driving lugs for use in field work, and a number of different types of such lugs are in use. The most familiar are angle iron lugs, which

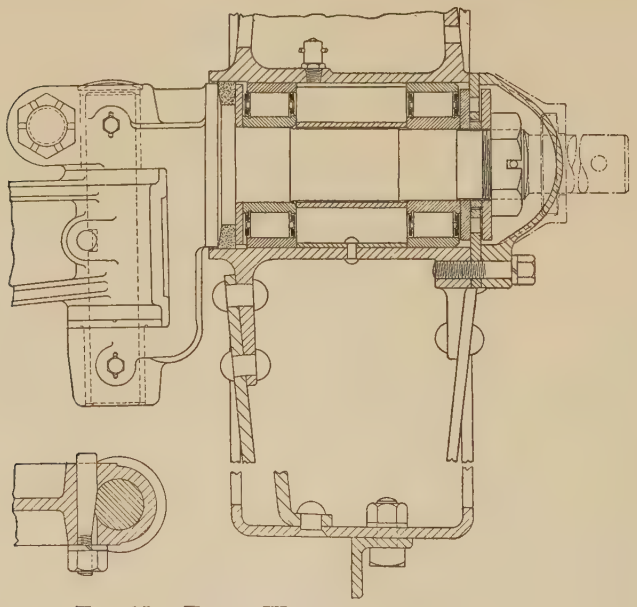


Fig. 15.—Front Wheel with Skid Ring.

are bolted to the rim of the wheel at an angle with its axis, and spade lugs, which are now often pressed of sheet steel. Both types are illustrated in Fig. 16.

Power Take-off—Farm tractors often are used to haul machinery (such as self-binders) that requires a source of power for operating it. When tractors were first used for this purpose a separate small engine was often provided, but later the power take-off was introduced, by means of which power from the tractor engine can be transmitted to the im-

plement. The power take-off usually consists of a pair of gears within the transmission housing of the tractor, operating a shaft extending lengthwise of the tractor either inside or outside of the housing, the rear end of the shaft being splined. A typical power take-off for operating farm implements is shown in Fig. 17. The driven gear ordinarily turns free on the shaft, but can be locked thereto by the positive clutch shown. It is, of course, very much cheaper to provide such a power take-off than to provide a small gasoline engine, and besides, the power take-off causes much less worry to the operator than would a small auxiliary engine on the implement.

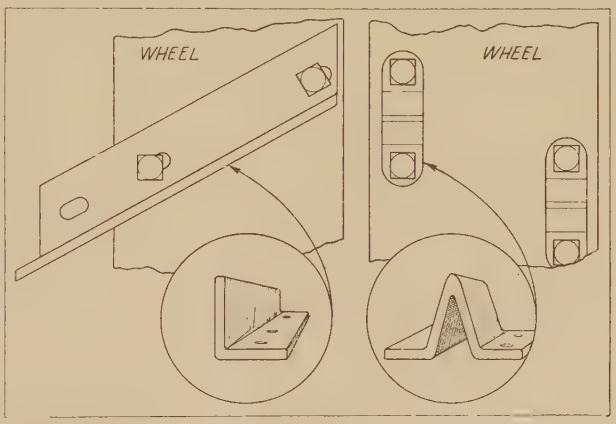


Fig. 16.—Angle Iron and Spade Lugs.

While some tractors do not include a driver's platform and fenders in their standard equipment, most tractors are furnished with these items as they leave the factory. A substantial rear platform is a safety feature, as it positively prevents the tractor from turning over backward; but its chief object is to enable the driver to sit and stand on the tractor at will, which adds greatly to his comfort. On wheeled tractors the so-called implement type of seat is generally furnished, which is pressed from sheet steel, and is supported either on a flat spring bolted to the top of the drive housing or on a coiled spring surrounding a vertical pillar on which it can slide. Crawler tractors, which on account of their smaller length are subjected to more jolting, are generally

furnished with a wagon-type of seat with cushion. A tool box can be readily mounted on the driver's platform.

Crawler Tractors—Crawler tractors were developed for use on soft ground where a wheeled tractor—even one with large wheels—would become mired. They find application in the cultivation of rice fields, in road building, in logging in swampy districts, etc. At first sight it would appear that a crawler tractor must be considerably less efficient than one of the wheeled type, for the reason that it must carry its

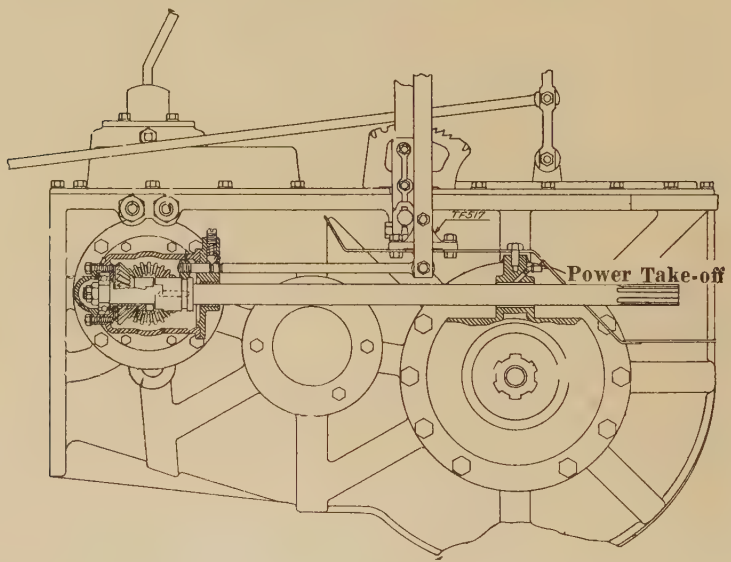


FIG. 17.—POWER TAKE-OFF.

tracks with it, which latter generally represent between 15% and 20% of the total weight. This is largely extra weight, which requires power to move it around the field. However, it is evident from the results of the Nebraska tests that, in spite of this fact, the crawler tractor is somewhat more efficient, on the average, than the wheeled type. By this is meant that it converts a larger proportion of the engine horse power into drawbar horse power.

There appears to be no reason why the transmission losses should be materially different, although, since the driving sprocket has a smaller pitch diameter than the outside diameter of the driving wheel of the wheeled tractor, with the

same engine and tractor speeds the transmission need not produce quite so large a speed reduction in the case of the crawler tractor. However, the chief reason for the higher efficiency of the latter is undoubtedly a reduction in the losses due to moving the tractor itself around the field.

Owing to its much greater supporting area, the crawler tractor sinks into the soil to a much lesser depth, hence it does less work in compacting the soil. The figure of 75% has been given as an average for the efficiencies of wheeled tractors, and 80% is a comparable figure for crawler tractors. However, the efficiencies of both wheeled and crawler tractors

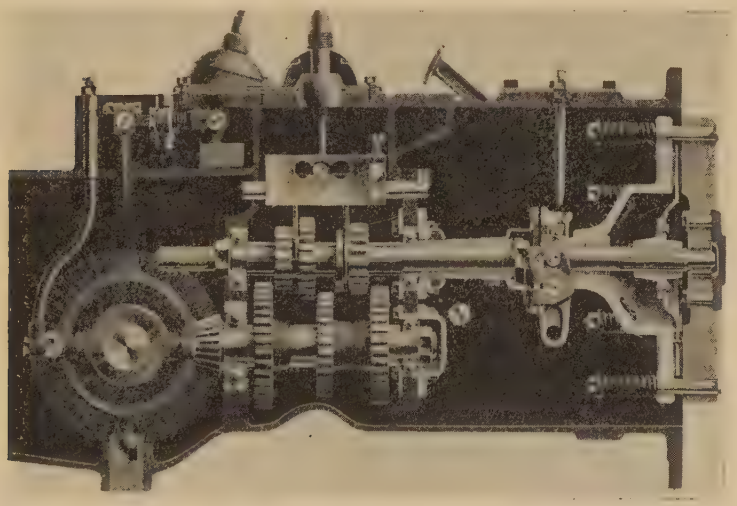


FIG. 18.—TRANSMISSION OF CATERPILLAR TRACTOR, IN VERTICAL SECTION.

vary over a considerable range, and in individual cases it is quite possible for a wheeled tractor to be more efficient than one of the crawler type.

The particular requirements which a tractor of the crawler type should meet may be briefly summarized as follows: It must

1. Distribute the weight of the machine over a large ground surface, and thus reduce ground depression, so as to be better able to operate over soft ground.

2. Distribute the tractive effort over a greater surface, and thus have the shear value of a greater area of soil.

3. Bridge depressions in the soil, and thus provide as smooth riding qualities as possible.

4. Be compact, with low center of mass and small width, making it suitable for all classes of operation.

5. Be capable of flexible control, making short turns and giving quick action in tight places.

6. Increase the efficiency of transmission from powerplant to drawbar.

General Arrangement—The general arrangement of the powerplant of a crawler tractor is very similar to that of a wheeled tractor, the engine, clutch, change gear and final reduction gear being combined in a single unit (see Fig. 18) with the crankshaft axis extending in the direction of motion of the tractor. The final drive shafts, which correspond to the driving axle shafts in the wheeled tractor that carry the drive wheels, in the case of the crawler tractor carry sprocket wheels with which the linked tracks of the tractor engage. There is this difference in the case of the crawler tractor, however, that provisions are made in it whereby all of the engine power can be transmitted to the driving sprocket on one side of the tractor and the driving sprocket on the opposite side held from rotation, at the will of the operator. When this is done, one of the tracks remains stationary and the tractor is moved around in a curve by the other track. It is by this means that a crawler-type tractor is steered.

The top surfaces of the track links form practically smooth steel tracks on which the weight of the tractor is supported through a number of track rollers. These rollers turn on pins supported on track frames which are made of castings, rolled channels or pressed steel members. The track frames generally are connected to the power unit in such a way that they can rock around the axis of the driving sprockets as a fulcrum, while the forward end of the power unit is supported upon them through a spring or springs.

In addition to passing over a rear driving sprocket, each track chain passes over a forward wheel, which may be either smooth or provided with teeth with which the rollers of the track chain engage. The distance between the axes of the driving sprocket and the front wheel is known as the chain center distance. This, however, is not a fixed distance, provisions being made for moving the forward wheel further away from the driving sprocket as the chain wears and thereby take up slack. The bearings for the front wheel, moreover, are not held rigidly in position, but are cushioned by a spring so as to relieve any shocks due to hard objects getting in between the track chain and sprocket.

Weight Distribution—In a crawler tractor all of the weight is at all times available for traction purposes, for all of it is supported on the track chains, and since the driving wheels are positively geared to the track chains, there can be slippage only if the tracks slip on the ground. From this point of view, therefore, it does not make any difference how the weight is distributed with respect to the contact area between track and ground. From another point of view, however, this is quite an important matter.

The same as in a wheeled tractor, when power is applied to the driving wheels (sprockets) the tractor tends to rise in front. Now, the object in using chain tracks instead of wheels is to get a larger supporting surface on the ground, and if all of the weight were removed from the forward end and concentrated at the rear end—by the driving force, which

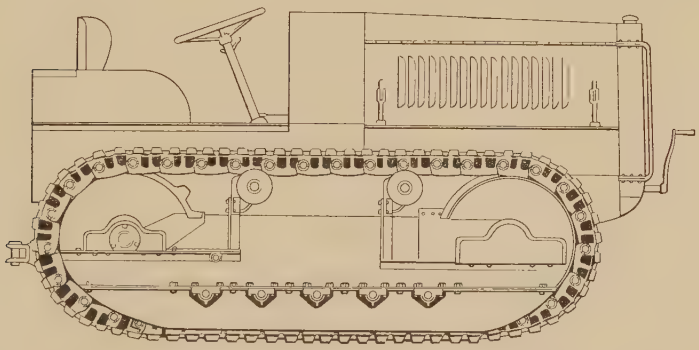


FIG. 19.—SIDE ELEVATION OF TYPICAL CRAWLER TRACTOR (Monarch).

tends to cause the tractor to turn around the driving sprocket axis—then the object aimed at by the use of chain tracks would be defeated; for if the weight is all concentrated on the rearmost supporting roller, then the track link on which that roller bears may sink into the soil as deeply as a wheel would. Hence the weight of a crawler tractor should be so distributed that when it is at rest and has no power applied, there is considerably more weight at the forward than at the rear end, and the weight becomes substantially equally distributed among the supporting rollers when the tractor is exerting its normal drawbar pull. This principle has gradually come to be recognized by tractor designers, and as a result the engines are now being placed farther forward with

respect to the tracks than was the case some years ago. Fig. 19 is a good illustration of present practice with respect to the general lay-out of crawler type tractors.

General Proportions—The average weight per drawbar horse power of crawler-type tractors now on the American market is about 300 lbs., but the most advanced designs with engines running at fairly high speeds weigh only about 250 lbs. per drawbar hp. Of this total weight the tracks usually constitute between 15% and 20%.

The center distance of the chain wheels, often referred to as the wheelbase, increases, of course, with the drawbar hp. It might be argued that since the maximum tractive force which the tractor can exert varies directly as its total weight, and since for a given speed the drawbar horse power is directly proportional to the total weight, which varies as the cube of the linear dimensions, the wheelbase should vary as the cube root of the drawbar horse power. However, the length, width, and height of tractors are not varied proportionally, and application of the cube-root formula to data at hand shows that it does not fit the case. As a matter of fact, the length is increased more rapidly than the width and height. Current practice with respect to crawler-tractor wheelbase lengths can be expressed with a satisfactory degree of accuracy by saying that the wheelbase is made equal to 40 in. plus one inch for every drawbar horse power. This gives 55 in. for the wheelbase length of a crawler tractor of 15 drawbar hp., and 65 in. for a 25 hp.

Track Pressures—In figuring the specific pressure of the tracks on the ground it is customary to assume that the bearing surface extends the whole length of the wheelbase, although this is not strictly correct. The method, however, may be used with satisfaction for comparative purposes. The product of the length of the wheelbase by the width of the track shoes then gives the supporting surface for each track, and dividing the total weight of the tractor by the combined surfaces of the two tracks then gives the specific track pressure. This usually varies between 5 and 6 lbs. per sq. in.

The effective bearing surface of the tracks is practically limited to that of the links on which the track rollers bear, because the lowest point of the chain wheels is at a higher level than the lowest part of the track wheels, and the track links which do not receive any direct pressure from the track rollers are inclined upwardly and do not press as hard against the ground as do those on which the rollers bear, hence they do not carry so much load—if they carry any at all. If only

the ground contact area of the links which receive the pressure of the track rollers is counted, the specific ground pressure figures out to an average of about 10 lbs. per sq. in. Thus from the required drawbar horse power the probable weight of the tractor may be estimated, and a suitable wheelbase chosen. These factors being determined, the necessary width of track shoe follows from the permissible specific ground pressure of 5–6 lbs. per sq. in.

The pitch of the track links is made such that there are from eight to 10 links to the wheelbase. The pitch diameters

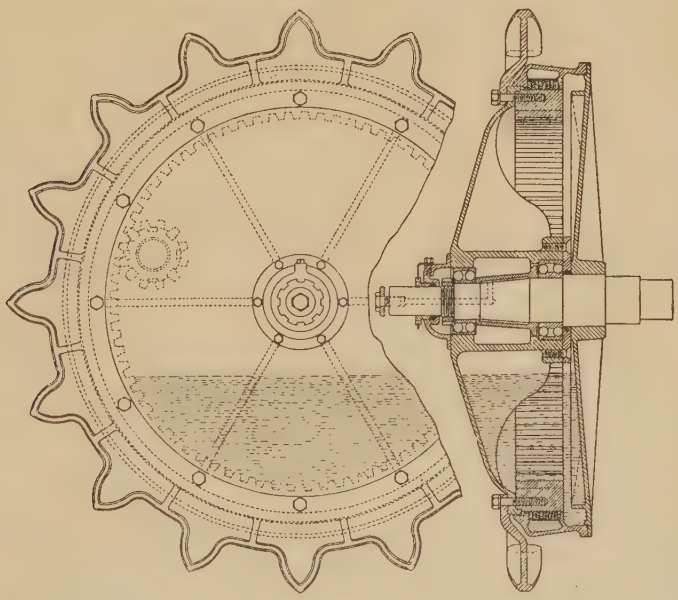


FIG. 20.—TRACK-DRIVING SPROCKET WITH INTERNAL-GEAR DRIVE.

of the driving sprockets are made equal to from 35% to 45% of the wheelbase. The approximate number of sprocket teeth can then be evaluated from the pitch and wheelbase chosen, and the exact pitch diameter required then calculated by the same method as used in calculating sprocket pitch diameters for use in connection with roller chains.

Sprocket Calculation—Sprockets for crawler tractors may have either the same pitch as the track, or one-half that pitch. These sprockets may be driven in at least three dif-

ferent ways. They may be secured to the ends of the differential shafts; they may be loose on axle spindles and driven by an internal gear and pinion set, or they may be loose on axle spindles and driven through pinions secured to the ends of differential shafts concentric with the axle spindles, intermediate gears and an internal gear ring. A sprocket driven by pinion and internal gear and having the same pitch as the track chain is shown in Fig. 20. If the pitch of the sprocket is the same as that of the track the number of teeth in the sprocket is usually less than 20, while otherwise it is over 20.

For sprockets with the same pitch as the track chain the necessary pitch diameter is found by means of the equation

$$D_p = \frac{P}{\sin (180°/N)}$$

where P is the pitch of the sprocket and N its number of teeth. If the sprocket has only half the pitch as the track chain, this same equation may be used, but P must then be made equal to the pitch of the chain and N equal to half the number of sprocket teeth. With sprockets of this type it is advantageous to use an uneven number of teeth, as in that case adjacent teeth will be loaded during alternate revolutions of the sprocket, whereas if the number of teeth were even, one half of the teeth would never carry any load and might therefore just as well be left off.

Track Links—The links of the track chain may be made either of castings in one piece, or they may be made up of drop forgings and rolled sections, the latter being used for the track shoes. If the links are cast, manganese steel is sometimes used, which is unusually hard as cast, and offers great resistance to wear. Adjacent links of the chain are joined together by means of pins and bushings, the same as in roller chains. Both the pins and the bushings are secured in the links, so they cannot rotate relative thereto, as these members usually have only a small bearing area in the links, and wear would be rapid if they were allowed to move therein. Motion, of course, takes place between each pin and its bushing as the chain curves around the sprocket. The bearing area of the pin in the bushing should be proportional to the maximum drawbar pull at plowing speed, and since the plowing speeds of different tractors do not vary greatly, it can be made proportional to the drawbar horse power. The area of the bearing surface is made equal to

about 0.3 sq. in. per drawbar horse power, and the ratio of length of bearing to diameter is made about 3.75. Pins and bushings are both case-hardened.

Figure 21 shows one design of track link that has been used by the Cleveland Tractor Company. The track shoe is slightly convex toward the outside and is cast integral with a transverse driving lug perpendicular to it. At each end of the shoe there is a pair of lugs for the link pin and bushing, those for the bushing being so spaced that they will enter between the pair for the pin. The pins are held from drifting and against angular motion by cross pins extending through holes in them near their ends and into slots machined in the outer ends of the lug bosses. These lugs are connected by the rail portions of the link which form a rolling surface parallel to the plane through the link pin axes.

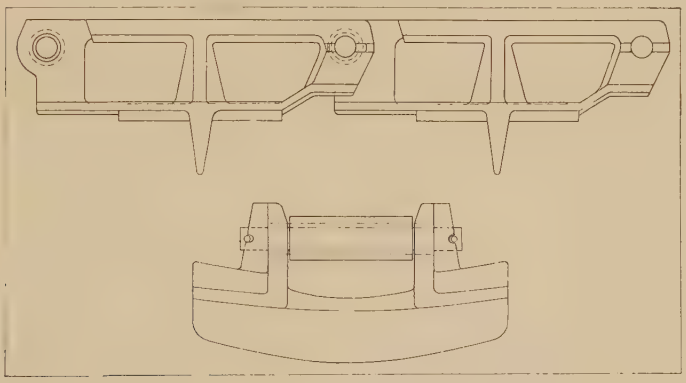


Fig. 21.—Cletrack Track Links.

A built-up track link is shown in Fig. 22 this design being used by the Caterpillar Tractor Company. The track shoes are cut from specially rolled steel bars and have the driving lug or grouser formed integral with them, the lug being of the same form (that is, transverse and perpendicular to the shoe) as in Fig. 21. Sometimes the driving lugs are made to extend across the shoe at an angle, the same as the usual angle iron lugs applied to the wheels of wheeled tractors, while they have also been given an arrow-head form.

A track-chain link, although it may be made as a single casting, consists essentially of four elements, namely, the rail portions on which the track rollers run, the lugs or eyes by

means of which adjacent links are joined together, the shoe which presents a large surface to the ground, and the grouser or driving lug, which digs into the ground and prevents slippage of the track. The ends of the rail portions, which form the eyes, are reduced in width, and the rails of adjacent links overlap, so there is no discontinuity of the rail surfaces on which the track rollers run. Between the two rail portions of each link there is a space into which the driving sprocket and possibly also the guide wheel rim extends, so the teeth on the sprockets may engage with the bushings or rollers below the top of the rail portion. A good rule for the necessary width of sprocket is "square root of maximum drawbar pull in plowing gear divided by 24."

Track Rollers—The tractor is supported upon the track by track rollers, which vary in number from three to seven.

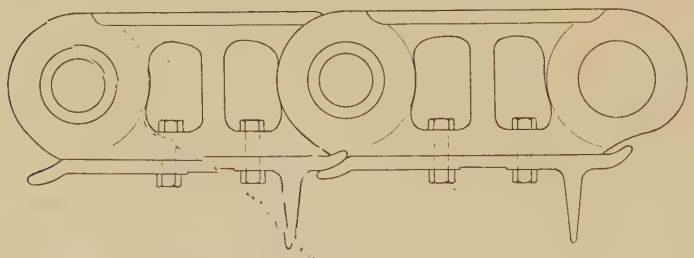


FIG. 22.—CATERPILLAR TRACK LINKS.

The diameters of these track rollers vary from about one-half to less than one-quarter the sprocket pitch diameter, their diameter being made smaller, relatively, the greater the number of rollers employed. Use of a comparatively large number of rollers has the advantage that it distributes the weight of the tractor over a larger number of track links. Ordinarily the track rollers are carried on spindles mounted in brackets on the track roller frame, but in some designs the spindles for the rollers are supported at the ends of levers. This arrangement permits of a more nearly uniform distribution of the load among the different rollers when traveling over uneven ground.

A sectional view of a track roller and its support, as used on Caterpillar tractors, is shown in Fig. 23. Each roller is mounted on two Hyatt flexible roller bearings on pins rigidly secured into brackets on the track frame. These pins are drilled out to form lubricant retainers, and they are provided with self-adjusting cork seals to keep the lubricant in. It

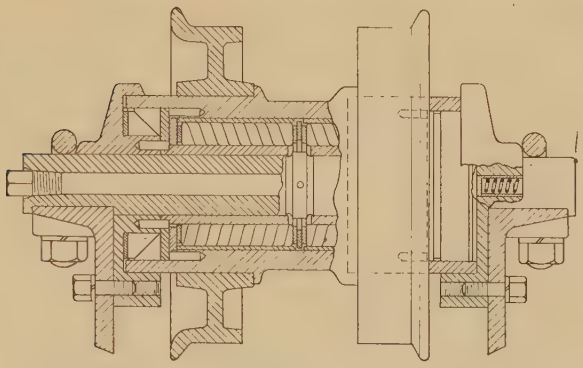


FIG. 23.—CATERPILLAR TRACK ROLLER IN SECTION.

will be noted that the track rollers are provided with flanges which straddle the track. In some designs both flanged and plain track rollers are used.

The upper section of the track chain in most cases is supported by track-carrier rollers, either one or two being used for each track. The design of the Caterpillar track-carrier roller is shown in Fig. 24. The space between the two members of the track frame is covered by means of a suitable sheet-metal "roof," so that any dirt which may drop off the upper section of the track cannot drop onto the lower section and cause trouble with the track rollers, sprockets and link pins.

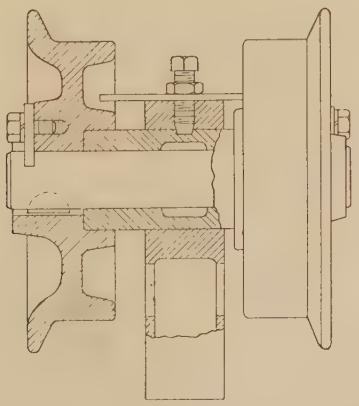


FIG. 24.—TRACK-CARRIER ROLLER IN SECTION.

Steering Gear—Crawler-type tractors are steered by applying power to the track on one side only. The way in which this is accomplished in the Caterpillar tractor is plainly shown in Fig. 25. The power is transmitted through the change gear to a bevel gear located at the middle of the tractor. From this bevel gear the power is transmitted to bull pinions on opposite sides through two multiple-disc-type friction

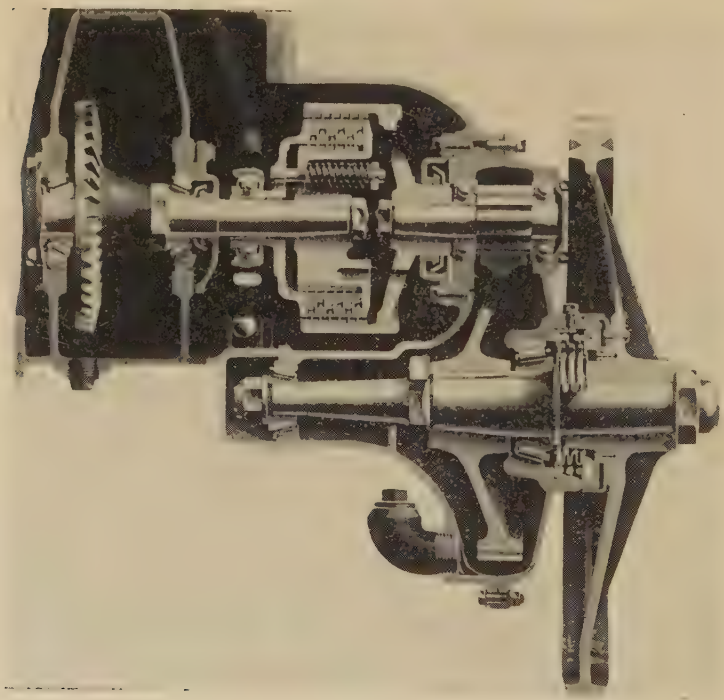


FIG. 25.—CATERPILLAR STEERING CLUTCHES AND OTHER PARTS.

clutches. Each of these clutches is controlled by means of a separate lever. By disengaging one clutch, all of the power of the engine is applied to one of the tracks, and this track naturally runs ahead of the other, causing the tractor to turn. If it is desired to make a very short turn, this can be accomplished by applying a brake to the drum of the clutch which is disengaged. There are two brakes, one acting on the outside of each steering clutch drum, each of them being controlled by a separate pedal. When the brake is applied to the drum of the disengaged clutch, the track on that side

is locked and the tractor turns around the center of that track as an axis. In this tractor the two large friction clutches and clutch brakes take the place of the usual steering gear, the differential gear and the ordinary single brake.

A somewhat different method of steering is employed on the Cletrac tractor, whose steering mechanism is illustrated in Fig. 26. Here the power is transmitted through a differential of the spur-gear type, to sprockets on opposite sides

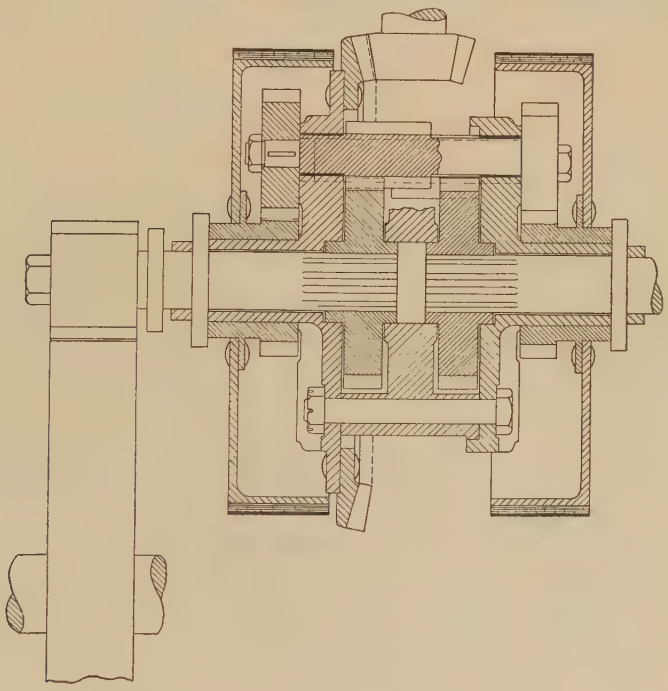


FIG. 26.—CLETRAC STEERING MECHANISM.

of the tractor. The differential is of the spur-gear type and the stud of one differential pinion of each pair is mounted in bearings in the differential housing and has a spur gear keyed to it at each end, which meshes with an equal-sized spur gear on the hub of the housing. These two spur gears on the hubs of the differential housing have brake drums secured to them. When one of these drums is being held from rotation by having the brake applied to it, the speed of the differential side gear (and of the track) on that side is decreased, while the speed of the differential side gear (and of the track) on the opposite side is increased. The tractor then describes a curve,

the "outside" track traveling about twice as fast as the "inside" one. These brakes are controlled by means of a steering wheel, and steering is therefore effected in the same manner as in a wheeled tractor.

Figure 27 shows a side view of the Caterpillar Ten tractor. The two levers by which the tractor is steered are plainly shown, as is also the method of mounting the bearings for the front sprocket wheel spindle slidingly on the track frame and placing the track chain under tension by means of heavy coiled springs, on opposite sides of the track frame. The track frame in this case is built up of rolled channels and the bearing spindles for the track rollers are supported below the channels, while that for the front sprocket is above them.

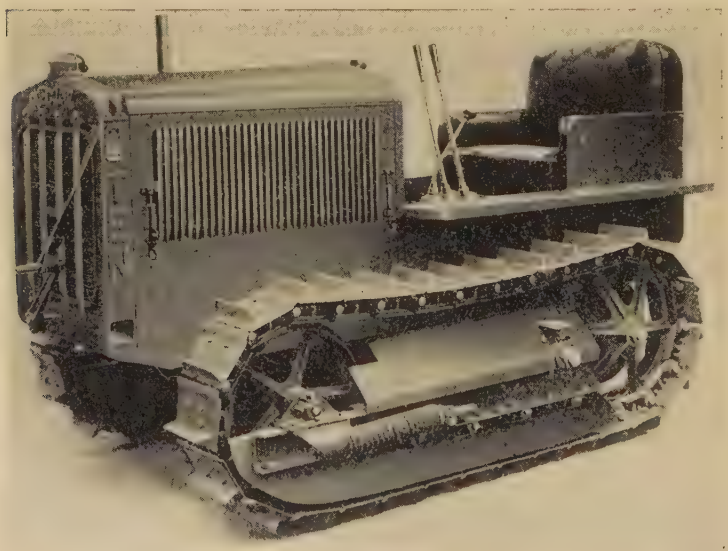


FIG. 27.—CATERPILLAR TRACTOR IN SIDE VIEW, SHOWING METHOD OF CUSHIONING GUIDE WHEEL.

CHAPTER XXIII

Fits and Tolerances

Motor vehicles are manufactured on the interchangeable plan; that is, all of the parts going into them are being produced continuously in large numbers, and from the production departments they are transferred to the assembly lines, where they must go together and give the proper fit without having any work done upon them there. If parts could be produced to specified dimensions with absolute accuracy, such interchangeable assembly would involve no difficulty, but that is impossible. Certain slight variations must be allowed in all dimensions of a part which have to conform to dimensions of other parts. Such variations are known as tolerances. Most of the surfaces with which we have to deal in automotive production and in machine construction generally, are turned or ground cylindrical surfaces—those of shafts and pins, and of the holes into which these shafts and pins must fit. Generally the most important dimension of such a surface, on which tolerances must be set, is the diameter. However, tolerances are required also on diameters of conical surfaces (for taper fits) and on lengths. These tolerances must be set by the engineering department and entered on the drawings, hence the designer must familiarize himself with the general principles underlying the determination of fits and tolerances.

Types of Fit—There are quite a number of different types of fit, from a free-running to a force or shrink fit. For a free fit of any kind the shaft must be of slightly smaller diameter than the hole, whereas for a force fit the shaft must be of larger diameter.

It is usual in machine design to make cylindrical parts (as well as others) to dimensions which can be expressed in round figures, say, even sixteenth of an inch. But since for most classes of fit there must be a slight difference between the nominal diameter of the shaft and that of the hole, it is obvious that the basic or nominal diameters of both cannot

be expressed in round figures. It is possible to make either the basic diameter of the shaft or the basic diameter of the hole a round figure, and this gives us two distinct systems. The system in which the shaft has a basic diameter expressed by a round figure is known as the basic-shaft system, and that in which the hole has a basic diameter expressed by a round figure is known as the basic-hole system. Each system possesses certain definite advantages, but it is generally agreed that for most purposes the basic-hole system is preferable.

Basic Shaft vs. Basic Hole—The basic shaft system has advantages where materials made to standard dimensions, such as cold-rolled shafting, are to be used. The maximum shaft diameter then is the basic or nominal size, and the shafts can be used without machining. The nominal bore in this case is varied in accordance with the fit required. The system is said to have advantages also in connection with ball bearings, which are made with a maximum bore of inner race and a maximum diameter of outer race equal to the nominal size (no plus tolerance). This system would ensure that the inner race would always be a light press fit, as required, whereas the outer race could be made either a press fit or a snug fit, as might be required.

Most countries that have done any standardization work along this line have adopted the basic-hole system. The chief reason for preferring the basic-hole to the basic-shaft system is that the hole is usually finished by means of a reamer and must also fit a mandrel, and if the bore is constant for all classes of fit, it saves a good deal on reamer and mandrel equipment required. The situation with respect to these alternate systems may be briefly summed up by saying that the basic-shaft system should be given the preferance wherever the use of standard finished material results in maximum economy, while the basic-hole system should be followed where the use of standard tools represents maximum economy.

The difference between the nominal diameters of the hole and the shaft is known as the allowance. The allowance is positive if the nominal diameter of the hole is larger than the nominal diameter of the shaft, and negative if the nominal diameter of the hole is smaller than the nominal diameter of the shaft. However, negative allowance is generally referred to as interference. English writers usually divide fits into three general types, viz., clearance fits, interference fits and transition fits.

Unilateral and Bilateral Tolerances—As already pointed out, parts cannot be made with absolute accuracy to the nominal dimensions, and a certain tolerance in dimensions

must be allowed. Variations from the nominal dimension may be allowed in both directions, as indicated by a specified dimension such as 1.250 ± 0.001 in. This is known as the bilateral system of tolerances, and was in wide use during the early years of interchangeable manufacturing. The other, the unilateral system, consists in allowing a tolerance on only one side of the nominal dimension. The bilateral system may be combined with either the basic-shaft system or the basic-hole system, and so may the unilateral system.

With the basic-hole system and unilateral tolerances, the nominal diameter is the minimum diameter of the hole, and the maximum diameter of the hole is larger than the nominal by the amount of the tolerance. With the basic-shaft system the nominal diameter is the maximum diameter of the shaft, and the minimum diameter is smaller than the nominal diameter by the amount of the tolerance.

Definitions of Terms—Following is a set of concise definitions of terms frequently employed in discussions of practices in interchangeable manufacturing:

Fit in general denotes the physical correlation of two parts joined together, and is characterized by the clearance or interference.

Clearance is the free space between bore and shaft.

Interference is the amount by which the diameter of the shaft to be inserted exceeds the bore.

Allowance is the difference between shaft diameter and bore; it is positive when the shaft diameter is less than the bore and negative when the shaft diameter is greater than the bore. Positive allowance is clearance; negative allowance, interference.

Maximum and minimum limits are the extreme dimensions between which the actual dimensions of the completed part must lie.

Tolerance is the difference between the greatest permissible (maximum) dimension and the smallest permissible (minimum) dimension of a piece of work.

Minimum clearance is the difference between the minimum bore and the maximum shaft diameter.

Maximum clearance is the difference between maximum bore and minimum shaft diameter.

Minimum interference is the difference between the minimum shaft diameter and the maximum bore.

Maximum interference is the difference between the maximum shaft diameter and minimum bore.

Nominal dimension is the measurement which indicates the size of the part.

Deviation is the difference between the actual and nominal dimension.

Qualities of fit are grades of workmanship according to which the precision of fits is graded.

Unilateral system is a system in which there are tolerances on one side of the nominal dimension only.

Bilateral system is a system in which there are tolerances on both sides of the nominal dimension.

The basic bore system is characterized by the following:

The bore does not change with a change in the class of fit, whereas the diameter of the shaft is made different according to the class of fit. The downward deviation of the bore is zero, therefore the zero line is the lower limit of the bore diameter. Shafts on which there are fits of different classes must have shoulders.

The basic shaft system is characterized by the following:

The shaft diameter does not change with a change in the class of fit, whereas the bore diameter is made different according to the class of fit. The upward deviation of the shaft is zero, hence the zero line is the maximum or upper limit of the shaft diameter.

With unilateral tolerances the allowance represents the condition of the tightest permissible fit, where the largest internal member is mated with the smallest external member.

American Standard—A Tentative American Standard on Tolerances, Allowances and Gauges for Metal Fits was approved by the American Standards Association in 1925. It is a basic-hole system with unilateral tolerances, and provides for eight different classes of fit, namely, loose, free, medium, snug, wringing, tight, medium force and force and shrink fits. Each class of fit is characterized by a definite tolerance on each of the mating parts, and a definite allowance for a certain range of nominal sizes. The tolerances and the allowance are both based on the mean diameter of the range.

Naturally, the tolerances and allowances must vary with the diameter and in the American Standard the tolerances vary as $a \sqrt[3]{D}$, where a is a coefficient which varies with the class of fit, and for some fits is different for the hole and the shaft, while D is the mean nominal diameter of the range to which the tolerance applies. In clearance fits, where the nominal diameter of the hole is larger than that of the shaft, the allowance varies as $b \sqrt[3]{D^2}$, where b is a coefficient differing with the fit. In interference fits, where the nominal diameter of the shaft is larger than that of the hole, the interference or negative allowance varies as $c\ D$, the coefficient c

varying with the class of fit. In transition fits the allowance is nil, the nominal diameter of the hole being the same as that of the shaft.

Classes of Fit—Applications of the various classes of fit in the American Tentative Standard and formulas for allowances and tolerances are given in the following:

Loose Fit: This fit provides for considerable freedom and embraces certain fits where accuracy is not essential. Formulas used are—Hole and shaft tolerances, $0.0025 \sqrt[3]{D}$; allowance, $0.0025 \sqrt[3]{D^2}$.

Free Fit: For running fits with speeds of 600 r. p. m. and over and journal pressures of 600 lbs. per sq. in. and over. Formulas—Hole and shaft tolerances, $0.0013 \sqrt[3]{D}$; allowance, $0.0014 \sqrt[3]{D^2}$.

Medium Fit. For running fits under 600 r. p. m. and with journal pressures of less than 600 lbs. per sq. in.; also for sliding fits and the more accurate machine tool and automotive parts. Formulas—Hole and shaft tolerances, $0.0008 \sqrt[3]{D}$; allowance, $0.0009 \sqrt[3]{D^2}$.

Snug Fit: This is the closest fit which can be assembled by hand, and necessitates work of considerable precision. It should be used when no perceptible shake is permissible and where moving parts are not intended to move freely under load. Formulas—Hole tolerance, $0.0006 \sqrt[3]{D}$; shaft tolerance, $0.0004 \sqrt[3]{D}$; allowance, 0.0000.

Wringing Fit: This is also known as the tunking fit, and is practically metal to metal. Assembly is usually selective and not interchangeable. Formulas—Hole tolerance, $0.0006 \sqrt[3]{D}$; shaft tolerance, $0.0004 \sqrt[3]{D}$; allowance, 0.0000.

(The average interference of metal given is the desired condition and must be obtained by selective assembly, that is, by mating large shafts in large holes and small shafts in small holes.)

Tight Fit: Light pressure is required to assemble these fits, and the parts are more or less permanently assembled, such as the fixed ends of studs for gears, pulleys, rocker arms, etc. These fits are used for drive fits in thin sections and extremely long fits in other sections, and also for shrink fits in very light sections. Used in automobile, ordnance and general machine manufacturing. Formulas—Hole and shaft tolerances, $0.0006 \sqrt[3]{D}$; Average interference of metal, $0.00025D$.

(The average interference of metal given is the desired

condition and must be obtained by selective assembly, that is, by mating large shafts in large holes and small shafts in small holes.)

Medium Force Fit: Considerable pressure is required to assemble these fits, and the parts are considered permanently assembled. These fits are used in fastening locomotive wheels, car wheels, armatures of dynamos and motors, and crank discs to their axles or shafts. They are also used for shrink fits in medium sections or long fits. These fits are the tightest

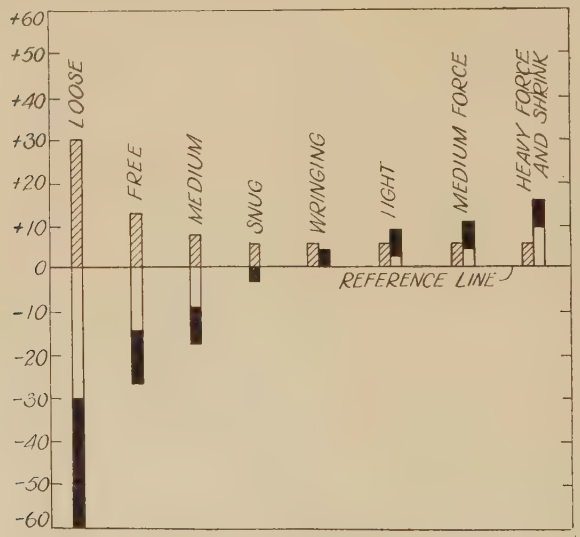


FIG. 1.—DIAGRAM OF AMERICAN TENTATIVE STANDARD FOR CYLINDRICAL FITS.

which are recommended for cast iron holes or external members, as they stress cast iron to its elastic limit. Formulas—

$$\text{Hole and shaft tolerances, } 0.0006\sqrt[3]{D};$$
$$\text{Average interference of metal, } 0.0005D.$$

(The average interference of metal given is the desired condition and must be obtained by selective assembly, that is, by mating large shafts in large holes and small shafts in small holes.)

Heavy Force or Shrink Fit: These fits are for steel holes where the metal can be stressed to its elastic limit; they cause excessive stress for cast iron holes. Shrink fits are

FIG. 2.—PLUG GAUGE.

used where heavy force fits are impractical, as on locomotive tires, heavy crank discs of large engines, etc. Formulas—

$$\text{Hole and shaft tolerances, } 0.0006\sqrt[3]{D};$$
$$\text{Average interference of metal, } 0.001D.$$

(The average interference of metal given by the equation is the desired condition and must be obtained by selective assembly, that is, by mating large shafts in large holes and small shafts in small holes.)

The Tentative American Standard as applied to a shaft with a nominal diameter of 1 in. is represented graphically in Fig. 1, allowances and tolerances being given in 0.0001-in. units. The tolerances for the holes are represented by cross-sectioned bars above the reference line, while the tolerances for shafts are represented by solid black bars. The allowance is represented by the space between the top of the solid black bar and the reference line.

Gauges and Gauging—Conformity with specified tolerances and allowances is assured by gauging. A gauge is a device for determining whether dimensions of a manufactured part are within specified limits. There is quite a variety of gauges, the principal types being ring, plug, snap and caliper gauges. A ring gauge (Fig. 4) is one whose inside measuring surfaces are either cylindrical or conical in form. A plug

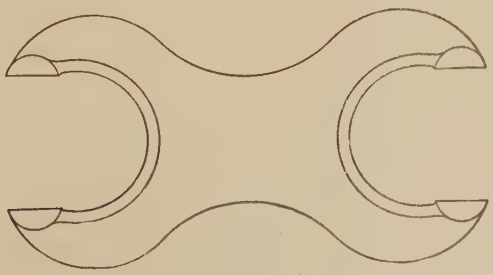


FIG. 3.—SNAP GAUGE.

gauge (Fig. 2) is one designed to verify conformity of holes with specified dimensions. Its measuring surfaces may consist of elements either parallel with or inclined toward an axis, while the closed figure represented by the intersection of the measuring surface with any plane perpendicular with the axis may be of any shape whatever. The cylindrical type of plug gauge, represented by Fig. 2, is the most common, however.

A snap gauge (Fig. 3) is one having inside measuring surfaces for calipering diameters, lengths, thicknesses, etc. A caliper gauge for internal members (Fig. 5) is similar to a snap gauge, while one for external members is similar to a plug gauge.

The system of gauging covered by the American Tentative Standard referred to in the foregoing is as follows:

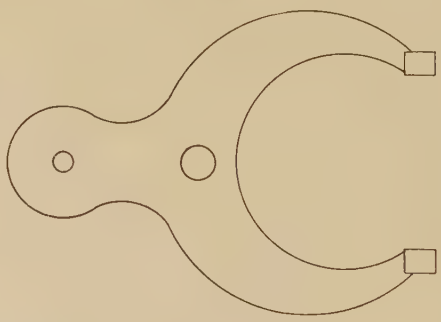


FIG. 4 (LEFT).—RING GAUGE.
FIG. 5 (RIGHT).—CALIPER GAUGE.

Direction of Tolerance on Gauges—The extreme sizes for all plain limit gauges shall not exceed the extreme limits of the part to be gauged. All variations in the gauges, whatever their cause or purpose, shall bring these gauges within these extreme limits. Thus a gauge which represents a minimum limit may be larger, but never smaller, than the minimum size specified for the part to be gauged, likewise the gauge which represents a maximum limit may be smaller, but never larger, than the maximum size specified for the part to be gauged.

The temperature of 68 deg. fahr. (20 deg. cent.) shall be the standard for plain limit gauges.

The final result sought by gauging is interchangeable manufacture in some degree. This means that the parts of a

mechanism can be assembled without *fitting* one part to another and when assembled the parts will function properly.

Principles of Gauging—Applied to manufactured material, the result sought is sufficient uniformity in size and contour to adapt the material without further fitting to the requirements of the industries. The fundamental principle involved in interchangeable manufacture requires that "a system of standardization and classification of fits shall establish a clearly defined line at which interference between mating parts begins." Hence,

(*a*) The standard or basic size, as physically represented by a correct standard master gauge, represents the line at which this interference begins between mating parts.

(*b*) It is the minimum size of the external members of all mating parts of standardized practice, regardless of the kind of fit.

(*c*) It is the maximum size of internal members of all mating parts where interference begins or that fit metal to metal.

The condition of (*b*) and (*c*) is represented by a mating pair of correctly fitting standard gauges.

(*d*) The limits of the component as physically represented by the limit master gauges shall not be exceeded as a result either of tolerance or wear of the gauges.

(*e*) "Go" gauges, or the equivalent verification of all the factors involved in the fit, are necessary to prevent interference of mating parts.

In the case of force fits, "go" gauges are necessary to determine the maximum amount of interference between mating parts.

(*f*) "Not go" gauges, or the equivalent verification of the determining factor are necessary to prevent the maximum looseness of mating parts exceeding the limits specified.

In the case of force fits, "not go" gauges are necessary to determine the minimum amount of interference between mating parts.

Tolerance for Automobile Parts—The following list of limits on parts of the Class B military truck may serve as a guide to the requirements in automotive work. These limits were established before the adoption of the Tentative American Standard on Tolerances, Allowances and Gauges and are based on a system of four fits as follows: No. 1, press fit; No. 2, running fit; No. 3, clearance fit; No. 4, shrink fit.

UNIVERSALS

Fit No.	Part				Material
1	Universal Yoke bushing	1.253	1.252	(grind)	1020 SAE
	Universal joint yoke—bore	1.2505	1.2495	(ream)	1040 SAE
2	Universal joint crossp in	.874	.873	(grind)	1020 SAE
	Universal yoke bushing bore	.876	.875	(grind)	1020 SAE
2	Distance between faces of cross	3.372	3.371	(grind)	1020 SAE
	Distance between faces of yoke	3.376	3.374		1040 SAE
2	Propeller shaft splines—top of keys	1.748	1.747	(grind)	1040 SAE
	Yoke keyways—bottom	1.750	1.749	(broached)	1040 SAE
3	Yoke keyways—top	1.422	1.417	(broached)	1040 SAE
	Propeller shaft splines keyway	(No bottom milled raidus)			
2	Propeller shaft splines—width	.271	.268	(grind)	1040 SAE
	Yoke keyways—width	.275	.273	(broached)	
3	Universal yoke stub shaft	2.267	2.264	(turn)	1040 SAE
	Universal dust cover	2.273	2.268		Steel stamping

FRONT AXLE

Fit No.	Part				Material
1	Steering knuckle pin bushing	1.8765	1.8755	(grind)	1020 SAE
	Front axle pin bushing bore	1.8750	1.8735	(ream)	1035 SAE
1	Cross rod yoke pin bush—upper	1.126	1.125	(grind)	1020 SAE
	Cross rod yoke bore—upper	1.125	1.124	(ream)	1025 SAE
3	Cross rod end pin—in arm	.884	.883	(grind)	1020 SAE
	Steering arm	.886	.884	(ream)	3130 SAE
2	Steering kunckle pin	1.497	1.496	(grind)	1020 SAE
	Steering knuckle pin bushing bore	1.500	1.499	(grind)	1020 SAE
2	Cross rod yoke pin	.884	.883	(grind)	1020 SAE
	Cross rod yoke pin bushing	.886	.885	(grind)	1020 SAE
3	Front hub—diameter	6.250	6.244		Malleable iron
	Front hub flange—bore	6.255	6.252		Malleable iron
	Camber of front wheels	2° toe in $\frac{7}{16}$ inch			

CLUTCH

Fit No.	Part				Material
2	Clutch shaft	1.497	1.495	(grind)	SAE 6120, 3120, 3320, 2320
	Brake flange	1.501	1.499	(ream)	1025 SAE
3	Clutch housing end plate	9.249	9.247		Aluminum SAE–31
	Clutch housing	9.252	9.250		Aluminum SAE–31
3	Clutch driving drum	11.500	11.496		Cylinder iron
	Flywheel drum recess	11.503	11.500		Cast iron SAE
2	Clutch driving disc	8.773	8.768	(pitch dia.)	CRS
	Clutch driving drum	8.805	8.800	(pitch dia.)	Cylinder iron

CLUTCH—*Continued*

Fit No.	Part				Material
2	Clutch driven drum	6.205	6.200	(O. D.)	Malleable iron
	Clutch driving disc	6.265	6.255	(inside dia.)	CRS
2	Clutch driving disc—O. D.	8.9687 (P. or M. .010)			
	Clutch driving drum—root of teeth	9.0364	9.0314		Cylinder iron
2	Clutch driven drum	5.999	5.994	(pitch dia.)	Malleable iron
	Clutch driven disc	6.023	6.018	(pitch dia.)	CRS
3	Clutch trunnion screw	.623	.621	(grind or polish)	CRS
	Clutch release brg. retainer hole	.626	.624	(ream)	Malleable iron
3	Clutch disc rivet hole	1560 (P. or M. .010)			
	Clutch disc rivet	.1560 (P. or M. .010)			Copper

SPRINGS

Fit No.	Part				Material
1	Front spring rear bracket bushing	1.129	1.127	(ream in place)	26 SAE
	Front spring rear bracket	1.125	1.124	(ream)	Steel casting
3	Front spring shackle bolt	.872	.871	(grind)	1020 SAE
	Front spring shackle bolt hole	.873	.872	(ream)	1025 SAE
3	Rear spring front bracket pin	1.243	1.241	(grind)	1020 SAE
	Rear spring front bracket	1.247	1.246	(ream)	Steel casting
1	Rear spring rear shackle bushing	2.004	2.002		26 SAE
	Rear spring shackle—rear	2.000	1.998	(ream)	Steel casting
3	Rear spring rear shackle pin	1.243	1.241	(grind)	1020 SAE
	Rear spring shackle pin bore	1.247	1.246	(ream)	Steel casting
2	Front spring shackle bolt	.872	.871	(grind)	1020 SAE
	Front spring eye bushing hole	.876	.872	(ream in place)	26 SAE
2	Rear spring shackle pin	1.243	1.241	(grind)	1020 SAE
	Rear spring front eye bushing hole	1.252	1.247	(ream in place)	26 SAE
2	Rear spring shackle bar	1.746	1.744	(grind)	1114 SAE
	Rear spring shackle bar bush hole	1.7500	1.7485	(ream in place)	26 SAE

TRANSMISSION

Fit No.	Part				Material
1	Transmission countershaft	2.1315	2.1305	(grind)	3320 SAE
	Countershaft gears	2.130	2.129	(grind)	2320, 3320, 2345, 3350 SAE
1	Idler gear bushing	1.628	1.627	(turn)	26 SAE
	Reverse idler gear	1.625	1.624	(grind)	2320, 3320, 2345, 3350 SAE
3	Reverse idler gear shaft	1.249	1.248	(grind)	2320 SAE
	Reverse idle gear shaft bore in case	1.250	1.249	(ream)	Aluminum SAE–31
3	Trans. main shaft—top of keys	2.245	2.240	(grind)	2320, 3320, 2345, 3350 SAE
	Trans. sliding gears—key bottoms	2.290	2.310	(broach)	2320, 3320, 2345, 3350 SAE
3	Trans. main shaft—bottom of keys	1.9105	1.9085	(grind)	2320, 3320, 2345, 3350 SAE
	Trans. sliding gears—key tops	1.913	1.912	(grind)	2320, 3320, 2345, 3350 SAE

TRANSMISSION—*Continued*

Fit No.	Part			Material
2	Trans. main shaft—key width	.556	.553 (grind)	2320, 3320, 2345, 3350 SAE
	Trans. sliding gears—keyway	.564	.560 (broach)	2320, 3320, 2345, 3350 SAE
2	Trans. shift forks—bottom jaws	2.770	2.760	1025, 1035 SAE
	Shift fork groove diameter	2.750	2.745	2320, 3320, 2345, 3350 SAE
2	Trans. shift fork—thickness	.433	.429	1025, 1035 SAE
	Trans. shift fork groove—width	.428	.436 (grind or polish)	
3	Sliding gear hubs	3.875	3.8735 (grind)	1020, 2345 SAE
	Sliding gear hub hole	3.862	3.860 (grind)	2320, 3320, 2345, 3350 SAE
1	Reverse idler gear shaft	1.247	1.246 (grind)	2320 SAE
	Reverse idler gear bush—bore	1.251	1.250 (ream)	26 SAE
3	Trans. shift fork bore	.861	.860 (ream)	1025, 1035 SAE
	Trans. shift fork rod	.8595	.85 5 (grind)	1020, 1010 SAE
3	Trans. front hanger—bore	6.006	6.004	
	Trans. front trunnion	6.000	5.998	

GEAR DATA

	Gear case centers	6.004	6.000
	First gear maximum pitch diam.	8.780	
	First gear minimum pitch diam.	8.798	
	Countershaft gear max. pitch diam.	3.198	
	Countershaft gear min. pitch diam.	3.184	
	Add other gears		

STEERING GEAR

Fit No.	Part			Material
	Jacket bushing—upper	1.872	1.871 (tin and sweat in place	26 SAE
	Jacket tube	1.875	1.873 (ream-steel tube)	16 B. W. G. (.065)
1	Upper housing cap bushing	1.877	1.876	26 SAE
	Upper housing cap bore	1.875	1.873 (ream)	Malleable iron
1	Pedal shaft bushing	1.253	1.251	26 SAE
	Pedal shaft bushing bore in cap	1.250	1.248 (ream)	Malleable iron
1	Steering column shaft bushing O.D.	2.3145	2.3135	26 SAE
	Housing bore—column bushing	2.3125	2.3105	Malleable iron
2	Steering column shaft	1.468	1.466 (grind) $\frac{3}{16}$ inch wall	Seamless steel tube
	Jacket bushing—upper	1.471	1.469 (ream)	26 SAE
3	Steering column shaft	1.479	1.477 (grind) $\frac{3}{16}$ inch wall	Seamless steel tube
	Steering column coupling	1.482	1.480 (ream)	DFS—malleable iron
3	Upper housing cap	2.9985	2.996	Malleable iron
	Main housing bore	3.000	2.9985 (ream)	Malleable iron
2	Worm shaft	1.5145	1.5135 (grind)	1020 SAE
	Upper housing cap bushing bore	1.516	1.515 (line ream)	26 SAE.
2	Pedal shaft	1.000	.998	CRS
	Pedal shaft bushing bore	1.003	1.001 (line ream)	26 SAE

STEERING GEAR—*Continued*

3	Steering gear housing cover (pr.)	6.124	6.122		Malleable iron
	Steering gear housing bore	6.125	6.123		Malleable iron
2	Worm wheel shaft	1.996	1.997	(grind)	1020 SAE
	Worm wheel bushings bore	1.999	2.000	(ream in place)	26 SAE
2	Spark tube bushing O. D.	.380	.378		Brass tubing
	Spark tube bore	.375	.374	(ream)	16 BWG steel tube
2	Throttle rod	.3135	.3115		CRS
	Spark tube bushing bore	.316	.315	(ream after press)	Brass tubing
2	Spark tube	.505	.495		Steel tubing
	Sector bracket	.510	.505	(ream)	Malleable iron
2	Worm gear thickness	1.625	1.622		
	Washers thickness	.187	.185		
	Steering gear depth housing	2.000	1.995		

Clearance with low limit case high limit gears = .002 tight.
Clearance with high limit case and low limit gears = .0105 loose.

REAR AXLE

Fit No.	Parts				Material
1	Housing sleeve (print dim.)	4.131	4.129	(selective fit)	3130 SAE
	Housing—bore (print dim.)	4.132	4.130	(15 ton pressure)	Cast steel
3	Housing sleeve	4.131	4.129		3130 SAE
	Brake anchor—bore	4.132	4.130		1020 or cast steel
3	Housing sleeve	4.123	4.124	(grind)	3130 SAE
	Housing sleeve bearing (roller)	4.125	4.126		
3	Axle shaft spline—top of keys	2.988	2.978	(driving flange was shrunk on the axle shaft)	3140 SAE
	Axle shaft drive flange—bottom of keyway	3.000	2.998		3130 SAE
3	Axle shaft spline—bottom	2.576	2.573		3140 SAE
	Drive flange top of keys	2.571	2.568		3130 SAE
4	Axle shaft spline key—width	.466	.464		3140 SAE
	Drive flange keyway—width	.464	.462		3130 SAE
3	Worm gear bore—spline top	11.650	11.645	(generated)	See page 1316
	Differential case tongue side—spline bottom	11.631	11.622	(generated)	1020 or steel casting
3	Worm gear bore—spline bottom	11.786	11.769	(generated)	See page 1316
	Differential case tongue side—spline top	11.758	11.755	(generated)	1020 or steel casting
3	Differential case recess side counterbore	10.376	10.375		1020 or steel casting
	Differential case tongue side fit in counterbore	10.376	10.375		1020 or steel casting
1	Differential case recess side fit in counterbore	11.768	11.766		1020 or steel casting
	Worm gear counterbore	11.766	11.764		See page 1316
	Differential case recess side width to shoulder	.7443	.7243		1020 or steel casting
	Differential case tongue side width	.752	.750		1020 or steel casting
	Worm gear hub thickness	1.530	1.500		See page 1316

REAR AXLE—*Continued*

Fit No.	Parts			Material
	Differential side gears and washers assembled in case	5.490	5.456	Make allowance of plus or minus for pinch fit of cases on gear
	Distance between faces in case	5.508	5.504	
3	Differential spider pin diameter	1.375	1.373 (grind)	2315 SAE
	Spider pin hole in housing	1.376	1.375	1020 or steel casting
2	Differential spider pin diameter	1.375	1.373 (grind)	2315 SAE
	Differential pinion—bore	1.379	1.377 (grind)	2315 SAE
2	Differential spider bore	4.255	4.253 (grind)	2315 SAE
	Differential side gear hub	4.249	4.247 (grind)	2315 SAE
3	Worm shaft diameter	2.5635	2.5630 (grind)	3120 SAE
	Worm shaft bearing diameter	2.5630	2.5640	
3	Axle shaft differential end—spline top	2.988	2.978 (grind)	3140 SAE
	Side gear splines—bottom	3.005	2.995 (broached)	2315 SAE
3	Axle shaft differential end—spline bottom	2.576	2.573	3140 SAE
	Side gear spline—key tops	2.581	2.579 (grind)	2315 SAE
2	Axle shaft differential end—key width	.466	.464	3140 SAE
	Side gear keyway—width	.471	.467	2315 SAE
3	Worm gear carrier step	15.995	15.998	Malleable iron
	Axle housing bore	16.000	15.994	Cast or pressed steel
1	Brake anchor pin (press)	1.127	1.126 (turn)	3120 SAE
	Brake anchor pin hole	1.125	1.124 (ream)	1025 or cast steel
3	Rear axle housing flange	9.252	9.249	Cast or pressed steel
	Brake anchor casting recess	9.257	9.253	1025 or cast steel
3	Rear wheel hub	7.998	7.992	Malleable iron
	Rear wheel hub flange bore	8.004	8.000	Cast steel
3	Rear axle shaft drive plate	9.250	9.245	3130 SAE
	Rear wheel hub flange—recess	9.257	9.252	Cast steel
2	Brake shaft	1.252	1.248	1020 SAE
	Brake shaft bushing—bore	1.262	1.259 (ream in place)	26 SAE
	Brakeshaft theoretical top of serration	1.219	1.209	1020 SAE
	Brakeshaft theoretical bottom of serration	1.141	1.131	1020 SAE
	Brakeshaft serration top	1.203	1.205	1020 SAE
	Brakeshaft serration bottom	1.145	1.147	
	Angle of serrations	9 degrees		
	Lever theoretical top of serration	1.143	1.141	1035 SAE
	Lever theoretical bottom of serration	1.221	1.219	1035 SAE
	Lever serration top	1.153	1.151	1035 SAE
	Lever serration bottom	1.211	1.209	1035 SAE
	Angle of serrations	9 degrees		
1	Brake shaft bushing (press)	1.502	1.501	26 SAE
	Brake shaft bushing bore in brake anchor	1.499	1.497	1025 or cast steel

CHAPTER XXIV

Methods of Rust-Protection

Cast iron and steels of various kinds form the bulk of the material entering into the construction of motor vehicles. Both of these are subject to corrosion when exposed to atmospheric influences, and in order to prevent the rapid deterioration of the vehicles, a good many of their parts must be suitably rust-proofed. It is fortunate that most of the working parts are constantly protected by coatings of oil or grease, so that their surfaces cannot be reached by the corrosive agencies in the atmosphere.

Corrosion is merely the oxidation of ferrous articles, and since there is oxygen in both the atmosphere and in water, it is not at all strange that iron should rust when exposed. However, the phenomenon of corrosion is not nearly so simple as appears at first sight. It has been found, for instance, that pure iron does not corrode when exposed to the atmosphere. No iron corrodes when exposed to dry air only, or when immersed in water that does not contain air. From these and other observations it has been deduced that the rusting of iron and steel articles is an electrolytic phenomenon. The impurities in the iron form with the iron an electrolytic couple, and any moisture on the iron or steel parts, either from a moist atmosphere or from wet roads, forms an electric conductor which closes the circuit, thus permitting an electric current to flow and chemical action to go on. Pure water has almost no electric conductivity, but the water is always contaminated and conducts current freely.

Much work has been done upon the improvement of rust-proofing processes since the advent of the automobile, and it is due to the success of these efforts that proportionally much less non-ferrous material is now being used in automobile construction than during the early years of the industry.

Paints and Enamels—Protective coatings applied to those steel parts of an automobile that are plainly in sight, such as

the body, hood, radiator shell fenders, wheels, etc., serve really a double purpose, in that they must not only protect the parts against rust but also must improve their appearance, and in many cases considerations of appearance have more influence in the selection of the finishing method than considerations of rust protection.

A very considerable number of different processes are available. There are, first of all, the paints, varnishes, lacquers and enamels that are applied to most of the larger exposed surfaces. As regards the sheet metal parts of the car, none are more exposed to corrosive agents than the fenders, filler plates and aprons, and these are generally finished in baked-on bituminous enamel. Two or more coats are applied, each coat being oven-baked at about 500° F. The enamel is applied by dipping, and the parts must previously be thoroughly cleaned, particularly of grease and rust, by passing them through baths of cleaning solution and pure water. The greatest care must be taken to have them absolutely dry before dipping in the enamel, as the latter will not be firmly bonded at any surfaces to which the least moisture adheres. Enameling is now generally carried on by the continuous process, the parts being hung onto a conveyor chain which is guided by a track that lowers the parts into a large tank containing the enamel, and then takes them through a drip channel and a drying oven, after which they are passed through another tank, drip channel and drying oven.

Metal Bath Coatings—Sheet steel is sold in the market which has coatings of non-corrosive metal applied to it by being dipped in baths of the molten metal. The most familiar of such rust-proofed sheet metals are galvanized (zinc-coated) sheet steel, tinplate and terne plate (lead coated). Both galvanized iron and terne plate are used for fuel tanks. Hot galvanizing is a quite effective method of rust-proofing, and that it is not used more in passenger cars is probably due to the fact that it does not give a very attractive finish. Sheet steel protected by one of these non-ferrous coatings lends itself particularly to use where soldering must be done. The coatings obtained by dipping are considerably heavier than those usually produced by that other method of applying surface layers of non-corrosive metals—electroplating. A hot-dip zinc coating will be of the order of 0.002 in. thickness, while electrically deposited coatings are from one-fifth to one-tenth as thick, and even less.

Nickel Plating—Nickel plating has long been used to give exposed metal parts an attractive finish, and probably saw

its first extensive use when the bicycle was at the zenith of its popularity. Its effectiveness as a rust preventive depends very much upon how the work is done. Electrolytically deposited coatings of nickel are always porous, and when nickel plating is used as a finish for steel articles, a coat of copper is usually applied first.

About 1923 the practice of finishing such exposed metal parts as the radiator shell, lamps and bumpers in nickel plate became very popular. However, in many cases the nickel-plated articles became rusty in a short time, and this led to a study of the process with a view to its improvement, by a number of large firms in the industry and by the Bureau of Standards. The following brief description of the process evolved by the Packard Motor Car Co. is indicative of the lines along which development has taken place with very satisfactory results.

Double Salt Solution—The Packard company uses a plating solution made up from ammonium sulphate and nickel sulphate, of a concentration of 4.85 oz. of nickel to the gallon. The solution is tested for strength once per week, and if it is found to be below the standard, salt in cloth bags is added. With this solution and a current density of 10 amperes per square foot, a uniform transfer of nickel from the metallic nickel anodes to the immersed parts is produced, under an electromotive force of 4–6 volts. As compared with the process formerly employed, both the concentration of the bath and the current density are higher, and this is said to give a harder deposit. The improved protection of the coating is indicated by the results of the salt-spray test, which will be described later on.

The current density employed in the copper plating which precedes the nickel plating was also increased, from 5 to 12 amperes per square foot. A double salt solution is used also in copper plating, the bath containing 5.1 oz. of copper cyanide and 6.5 oz. of sodium cyanide per gallon. In order to obtain a smooth nickeled surface, the surface of the base metal on which the nickel is deposited must be smooth. The surfaces therefore are buffed previous to plating, but steel is comparatively hard and it is impossible to obtain as smooth a surface as desired by buffing. In this connection the preliminary coating of copper has a very beneficial effect. Being soft, it flows under the action of the buffing wheel, filling out depressions in the surface of the steel and giving a smoother surface than that on which it was deposited.

List of Operations—One of the most exacting plating jobs is that on the radiator shell, and in the Packard plant the finishing of the shell includes 23 successive operations, as follows:

1. Polish (this involves really five distinct operations, with different wheels and abrasives).
2. Clean by immersion in potash solution.
3. Wash off in cold water.
4. Dip in 20% muriatic acid solution.
5. Dip in cold water.
6. Dip in potassium cyanide solution.
7. Dip in cold water.
8. Bench-brush with pumice and water, using electro-platers brush.
9. Dip in cold water.
10. Copper-plate in copper-cyanide solution for 30 minutes.
11. Dip in cold water.
12. Dip in hot water to accelerate drying.
13. Buff.
14. Dip in potash solution.
15. Dip in cold water.
16. Dip in potassium cyanide solution.
17. Dip in cold water.
18. Bench-brush corners at radiator water inlet.
19. Dip in cold water.
20. Nickel-plate in nickel-ammonium sulphate solution.
21. Dip in cold water.
22. Dip in hot water to accelerate drying.
23. Buff.

Thickness of Coating—Wm. Blum, who conducted research on nickel plating at the Bureau of Standards, says that the coats of nickel generally vary between 0.0001 and 0.0002 in. in thickness. He recommends that the coat of copper underneath be made relatively heavy, say, 0.0003 in., as the additional cost is more than made up by reduction in the time required for buffing in preparation for the nickel coat. The thickness of the coating of nickel and copper may be determined as follows:

Place a piece of paper 1 in. square on the surface of the plated specimen and coat the entire surface of the specimen with paraffin. Then remove the paper from the paraffin coating with a sharp knife and dissolve the plate below it with

nitric acid. The amount of plate metal in the acid is then determined by the usual methods of quantitative analysis.

Chromium-plating—In 1928 chromium plating largely superseded nickel-plating as a finish for the exposed bright parts of the car. The parts in most instances are still nickel-plated, but a thin coat of chromium is superposed on the nickel. Nickel is not very satisfactory as a finishing coat, because it tarnishes readily, and a plated surface cannot be restored by polishing, as the thin coatings would soon wear out and cause the base metal to rust. As a rust preventative chromium is no better than nickel, if as good, but it has two valuable qualities, that of being non-tarnishing in the atmosphere, and that of being very hard, so it cannot be injured by scratching. As regards the hardness of chromium plate, Wm. Blum, of the Bureau of Standards cites a case where a test with a Bierbaum scratch hardness tester under a certain load on the sapphire point produced a scratch 0.7 microns wide on the chrome plate, and a scratch 2.2 microns wide on the base metal of cold-rolled steel. (1 micron = 0.001 mm.).

Many of the earlier attempts to deposit chromium onto other metals from a chromic acid bath resulted in failure, because the experimenters did not use sufficiently high current densities. Chromium has comparatively little throwing power, especially if the current density is low. One of the pioneers in chromium plating was Prof. Sargent of Cornell University, who in 1920 introduced the chromic acid bath. This solution is generally used today, in a concentration of 33 oz. of chromic acid per gallon, to which from 0.4 to 0.7 oz. of chromium sulphate is added. Development work in connection with chromium plating in recent years has been devoted largely to the discovery of additive elements or catalysts which should increase the throwing power.

Temperature and current density are important factors in chromium plating, upon which the character of the deposit depends. With current densities of between 100 and 200 amperes per sq. ft. and a temperature of the bath of 112° F. a bright deposit is produced. But, as already stated, the chromium has poor throwing power, and in the case of deeply recessed surfaces particularly it is difficult to get uniform coats. There may be no deposit whatever at the bottoms of the recesses, while at the projecting portions of the surfaces the deposit may be gray and spongy. This is due to the fact that the cathode efficiency decreases rapidly with the current density. If the cathode has an irregular surface, the current density will be least at those portions of its surface which are most remote from the anode. According to Blum, at

112° F. the cathode efficiency is 18% for a current density of 200 emperes per sq. ft., and only 7% for a current density of 50 emperes per sq. ft.

With articles of steel the difficulty of obtaining a uniform coat is reduced by first depositing a coat of copper on the steel. Irregularities in the deposit are due to the overvoltage of hydrogen at the cathode with respect to the cathode metal, and hydrogen has less overvoltage relative to copper than to steel.

In addition to being used for external parts of cars, chromium plating is applied also to certain wearing parts, such as the piston pins of engines, which are subjected to the corrosive influences of moisture and sulphur in the crank-case due to fuel particles and burnt gases passing the pistons. Water pump shafts also have been chromium-plated to prevent their corrosion and rapid wear, but in one case at least with not altogether satisfactory results.

Cadmium-plating—Cadmium is a metal quite similar in its properties to zinc, and when applied to the surfaces of iron and steel articles it is a more effective rust preventative than the latter. Cadmium is slightly softer than zinc but harder than tin. It resembles silver in color and takes a very nice polish, which is not readily influenced by atmospheric conditions. An incidental advantage is that cadmium sulphide, which may be formed as a result of sulphur fumes in the atmosphere, has very little color.

While deposits of the metal can be obtained from both acid and alkaline baths, the latter give by far the better results and are used exclusively for plating. According to M. J. Dubbs, who presented a paper on Cadmium Plating before the Electroplaters and Depositors Technical Society (Great Britain), the plating bath is prepared from a double solution of cyanide of cadmium and cyanide of potassium, the addition of a little hydrate of sodium and dextrine facilitating the deposition and rendering the coating smoother, thus permitting of a better finish.

With a solution composed of 3 lbs. of double cyanide of cadmium and potassium, 0.12 to 0.24 lb. of sodium hydrate, and 400 grains of dextrine to 4 U. S. gallons, used cold, a current density of 14 amperes per sq. ft. at 2 volts is called for.

From various tests it appears that the cadmium combines with the iron even when cold, but this action may be greatly accelerated if the plated article is maintained at a temperature of 400° F. in a non-oxidizing atmosphere for some hours. Pieces thus treated do not show any rust when exposed to

the atmosphere continuously for one year, even though after a polishing operation all traces of cadmium seem to disappear.

In the United States the Udylite Company started a cadmium plating business in 1919. Its process has been extensively used for rust-proofing automobile parts, such as bumper and chassis bolts, brake parts, etc. The bath used has been constantly improved, especially with regard to its throwing power, and coatings are now obtained which are fairly uniform in thickness and quite adherent, so they will not lift when the base metal is flexed and twisted. As compared with zinc coatings, it is claimed that the cadmium plate is more nearly uniform, more continuous and more nearly free from pin holes.

The thickness of the coatings varies with the service for which the parts are to be employed. Perhaps one of the hardest services to which plated surfaces are exposed is that on cap nuts of demountable wheels. These are given coatings 0.0006 in thickness, while parts that are not subjected to abrasion are plated only to a depth of 0.0002 in. It is claimed that test specimens with a coating of cadmium plate applied electrolytically showed no rust after being exposed to the salt spray (20% solution) for 1860 hours.

That cadmium plating has attained considerable importance is shown by Government figures, according to which the industries of the United States in 1926 consumed 800,000 lbs. of cadmium as metal and sulphite, and that of this amount 131,736 lbs. was used for plating.

Sherardizing—This process, invented by Sherard Cowper-Coles, a British metallurgist, consists in packing the parts to be coated in powdered zinc of extreme fineness, heating the drums or retorts containing the parts at a slow rate to a temperature of 600° F., and then cooling them gradually. It is claimed that this treatment results in the penetration of the superficial structure of the steel, the result being an intimate mixture of the two metals in the surface layer, amounting almost to an alloying action. The parts so treated are said to be entirely free from defects sometimes found in galvanized parts. The coating will not peel or chip off under the hardest usage, and the physical properties of the metal treated are not adversely affected by the process, as they are said to be by galvanizing.

Vitreous Enamel—The rust-proofing of exhaust manifolds presents a problem of particular difficulty, as these manifolds in service sometimes reach a temperature corresponding to red heat. They are now given a vitreous enamel finish by several manufacturers, which has proven very satisfactory.

It is said that the coating of vitreous enamel will break down if the engine is operated on the test stand under full throttle at high speed for long periods, but in service on passenger cars the temperature of the manifold does not reach the break-down point of the enamel.

In applying this coating to the manifolds, after the latter have been thoroughly cleaned and dried, they are sprayed with powdered glass. After the coating has dried the manifolds are placed in a muffle furnace in which they are raised to a temperature of about 1500° F., which melts the glass and produces an impervious coating not affected by the highest temperatures reached by the manifold in automobile operation. The time of heating in the muffle varies with the thickness of the coat, being usually 12 minutes.

Parkerizing—Parkerizing consists in forming a layer of iron phosphates on iron and steel articles by immersing them in a bath of phosphoric acid solution to which certain other ingredients of the nature of catalysts or accelerators have been added, the bath being maintained at the boiling temperature. As with other processes, the surfaces of the parts to be treated must be perfectly clean, and especially free from grease and rust, as otherwise the protecting coating cannot be formed. This requirement can be met by treatment with any of the common cleaning solutions, as well as by sand-blasting.

When the parts are immersed in the solution, there is at first a rapid evolution of gas, which gradually slows down and ceases at the end of an hour or an hour and a half. The reaction is then complete, and the articles are removed from the bath and allowed to dry; and since they are at about 200° as they leave the bath, drying proceeds rapidly. Upon leaving the bath the parts have a grayish surface, which microscopic examination shows to consist of a dense layer of iron phosphate crystals. The parts are then subjected to a finishing process which varies with the service for which they are intended, and sometimes adds further to their resistance to corrosion. The usual finish is a mat gray.

It is claimed for the parkerizing process that, since the temperature to which the parts are exposed is limited to 212°, there can be no deterioration in the structure of the material; and since no outside material is deposited, there will be no change in fits, which is particularly important in the case of precision threaded parts. It is also claimed that a parkerized surface forms a very good foundation for coatings of paint, varnish and lacquer, and that lacquer adheres to it so well that the usual priming coats can be omitted.

Browning—A process which gives a slight amount of rust protection and is used chiefly—in the small arms and similar industries—because of the attractive appearance which it imparts to steel parts, is that known as "browning." The color imparted to the finished steel surface by this process is bluish-black, and the term "browning" therefore is somewhat of a misnomer.

The articles are first thoroughly cleaned in a soda solution with a concentration of from 5% to 15%. They are then dried in the atmosphere, and the browning solution is applied by means of a brush or sponge or by dipping. Quite a number of different browning solutions are in use; one of the most popular has the following composition: Perchloride of iron, 3.9%; bichloride of mercury, 1.2%; copper sulphate, 0.6%; nitric acid, 1.8%; spirits of niter, 6.2%; alcohol, 4%; water, 82.3%.

After the browning solution has been applied the parts are allowed to dry in the atmosphere for 30 minutes. They are then placed for 30 minutes in a drying room, which is maintained at a temperature of 170° F. From this room they are transferred to a humid room, which is maintained at a temperature of 160° F., and the object in keeping them in the drying room is to bring them up to the temperature of the humid room, so there will be no deposition of moisture on them when they are introduced in the humid room, in which the humidity is maintained at between 60% and 70% of the saturation point. Under the combined influence of the chemicals in the browning solution and the high moisture content of the atmosphere, a layer of oxides forms rapidly on the surface of the steel articles, which are kept in the humid room for 1½ hours.

At the end of this period the parts are boiled in clear water for 15 minutes, dried and carded. They are then again given a coating of the browning solution, dried, heated in the dry and humid rooms, and boiled, and this process is repeated twice more, hence the articles receive five successive coats of the solution.

Schoop Metal Spray—The Schoop metal spraying process, which is being exploited in this country by the Metal Coating Company of Cleveland, is not primarily a rust-proofing process, as it permits of applying coatings of almost any metal to both metallic and non-metallic articles, but it can be and is being used for rust-proofing purposes. In applying the process, use is made of a "gun" weighing 3½ lbs. Standard gauge wire of any metal is fed automatically to the nozzle of the short barrel, where it enters a continuous reducing

flame of gas, hydrogen, acetylene, etc. When the metal is melted it is blown by air under pressure, which issues from a concentric opening of the nozzle, onto the articles to be coated.

The process has been used to coat the iron protecting tubes of thermo-couples with aluminum, to prevent their oxidation at the high temperatures to which they are exposed. Attempts have been made also to coat exhaust manifolds with aluminum instead of with vitreous enamel, and the process is used to a certain extent for automobile repair work.

To obtain a good bond between the object coated and the coat, it is necessary that the former be thoroughly clean and of an open nature, so as to give a key for the deposit. Sand blasting with sharp sand has been found to give good results, while blasting with shot gives a surface that is too highly polished.

Salt-Spray Test—The effectiveness of rust-proofing processes is usually determined by means of the salt-spray test. The articles or specimens to which the process has been applied are suspended in a cabinet, at the bottom of which there is a quantity of salt solution, a 20% solution being usually employed. By means of an atomizer of the domestic type, salt solution is constantly sprayed into the air in the cabinet, but in such manner that it does not strike the rust-proofed surfaces directly. In this way a salty mist is maintained in the cabinet. There should be a thin film of the solution on the articles under test at all times, but no drops of sensible size.

Owing to the slight over-pressure produced in the cabinet by the atomizer, which is operated by air under light pressure, it is impossible to make the cabinet air-tight, but the breathing openings provided should be made no larger than required to relieve the over-pressure due to this cause.

When the coating starts to break down, red iron rust develops, usually in small pin points, at first, which extend gradually. By noting the character of the final breakdown, a good idea of the evenness of the coating may be obtained. In order to obtain results that are comparable, certain precautions must be taken, and one thing to look out for is that no oil gets into the cabinet (as from the small compressor furnishing the compressed air for the spray).

J. A. Kapp, who described the salt spray test before the American Society for Testing Materials in 1914, stated that a relatively thin protective coating breaks down under this test usually in 2 to 3 days; a well-applied coating will last at least a week, and if no rust appears in two weeks it may be assumed that rust protection will be assured under normal

conditions for an indefinite period. A sub-committee of the Committee on Corrosion of non-ferrous metals and alloys of the A. S. T. M. has been studying the salt-spray test for some years, and it is planned to evolve a standard method of applying the test.

Ferroxyl Test—The freedom from pin-holes of plated coatings can be determined by the ferroxyl test, which is made as follows: A hot solution containing potassium ferrocyanide, sodium chloride and agar-agar is flowed in a thin layer over the surface to be tested. As it cools, this solution quickly hardens to a jelly. The sodium chloride reacts with the iron exposed at the pin holes. A little iron goes into solution and reacts with the potassium ferrocyanide, forming a deep-blue coloration in the immediate vicinity of the pin-hole. The blue substance is probably Prussian blue, a mixture of a number of ferrocyanides. If a permanent record is required the layer of jelly is removed intact to glass or paper, and dried.

An improvement of this test, to better adapt it for use in the shop, has been suggested by Karl Pitschner. The reagent is sprayed on paper, which is dried in an oven and can then be kept indefinitely without deterioration. Paper so prepared is moistened on the sensitized side and spread over the dry surface to be tested, where it is allowed to remain for several minutes. After the paper is removed, all of the reagent may be washed out with water, leaving only blue spots indicating the pin holes. When operating in this way it is desirable to use a solution with from 50 to 100% higher concentration than where the solution is applied to the specimen directly.

APPENDIX

HARDNESS AND HARDNESS TESTS

Hardness is defined as the ability of a material to resist penetration or to penetrate other materials. The first attempt to obtain a means of gauging and expressing the hardnesses of different materials consisted in the establishment of the Mohs scale, a list of ten different materials, starting with the diamond, each of which would scratch the material directly below it in the scale, but could not be scratched by it. For certain lines of work a scratch test for hardness is still being used, the hardness being taken to be proportional to the pressure which must be applied to a sapphire point in order to produce a scratch of given width (as measured by means of a microscope) when drawn across the surface of the material under test.

Brinell Hardness Test.—A widely used method for testing materials for hardness is that due to the Swedish metallurgical Brinell. It consists in pressing a hardened steel ball into the surface of the material under test, under a known pressure, and measuring the diameter of the indentation made. A steel ball of 10 mm. diameter is used, and pressed into the surface of the material under a pressure of 3000 kg. (alternately a ball of 5 mm. diameter under 750 kg. pressure). The pressure on the ball, in kg. divided by the area of the spherical surface of the impression in sq. mm. is known as the Brinell hardness number. The test is applied by means of a press provided with a pressure gauge, and the diameter of the impression is measured by means of a measuring microscope furnished with the hardness-testing machine. At around 300 Brinell, which is characteristic of oil-hardened alloy steel, a difference of one point Brinell hardness corresponds to a difference in the diameter of the impression of only 0.00022 in., hence the measuring device must be very sensitive. After the diameter of the impression has been measured the corresponding Brinell hardness number is determined from a table.

For an accurate test the specimen is ground and polished at the point where the impression is to be made, and the load is applied gradually and held for at least 30 seconds. Two measurements are made of the diameter of the impression, at right angles to each other, and the mean of the two readings is taken. The test is not suitable for soft steel specimens less than ½ in. thick, and must not be made sufficiently close to an edge of the specimen to cause deformation of the edge as a result of a flow of the metal from under the ball.

Relation to Tensile Strength.—Hardness measurements are made in the automotive industry as a check on the quality and uniformity of materials, and on the effectiveness of heat treatments applied. For materials of a given class the hardness number is a fair index of the machining qualities, and it is also an index of the tensile properties. A regular tensile test can be made only in a quite expensive machine

on a specially prepared test specimen, whereas a Brinell hardness test can be made on a less costly machine, in considerably less time and often directly on production parts without destroying the latter, and the results obtained as regards tensile properties are sufficiently accurate for all practical purposes. Although the Brinell is an empirical test and there is no absolute relation between the Brinell hardness number and the ultimate strength as measured in a tensile testing machine, still the proportionality is surprisingly close for all kinds of steel.

The British Engineering Standards Association recommends a ratio of 493 (lbs. p. sq. in. tensile strength per point Brinell hardness) as a mean value for all steels. The German Engineering Standards Committee recommends a ratio of 515 for carbon steels of between 82 and 276 Brinell hardness and a ratio of 470 for alloy steels of between 178 and 276 Brinell hardness. R. H. Greaves and J. H. Jones of the Woolwich (England) Arsenal research department, in a paper presented to the Iron and Steel Institute in 1926, suggested the following ratios, based upon a large amount of data on hardness and tensile tests on the same specimens obtained at the arsenal and elsewhere:

For heat-treated alloy steels of 250–400 Brinell	470
For heat-treated alloy steels below 250 Brinell	480
For heat-treated carbon steels	480
For medium carbon steels, rolled, normalized or annealed	493
For mild steels	515

Rockwell Hardness Test.—The Rockwell hardness test is similar in principle to the Brinell, in that an indentation is made in the specimen by a hardened steel (or diamond) penetrator of standard form under a standard pressure, but in the Rockwell test the depth of the impression is measured instead of the diameter, which is advantageous from the standpoint of convenience. After the specimen has been placed in the tester it is elevated to the testing position, whereby an initial small load of 10 kg. is automatically applied to the penetrator. This serves to break through any light scale or coating and to bring the penetrator in direct contact with the material of the specimen. The direct-reading dial gauge of the tester is then set to the zero position.

Next the major load of either 100 or 150 kg. is applied to the penetrator, forcing the latter into the surface of the specimen. While the major load is on, the gauge indicates not only the depth of penetration but also the deformation or elastic yield of the press, and readings are therefore taken only after the major load has been removed. The scale reading then gives the Rockwell hardness number directly.

Two types of penetrator, with a separate load for each, are used with the Rockwell hardness tester, and the dial of the instrument is therefore provided with two scales. One of the penetrators is a diamond cone with 120° conical angle and a "point" ground to a definite radius, which is forced into the specimen under a pressure of 150 kg. With this penetrator and load, which are used on the harder metals, the C scale is used. The other penetrator is a $\frac{1}{16}$-in. hardened steel ball, which is forced into the surface of the specimen under a load of 100 kg. With this the B scale is used. In giving the Rockwell hardness of a material it is therefore important to always mention the scale to which the figure refers. The same as with the Brinell method, certain precautions are necessary if accurate results are required.

Shore Scleroscope.—A hardness tester based on an entirely differ-

ent principle from that of the two instruments already described is the Shore scleroscope. A diamond-pointed hammer or tup is dropped onto the specimen from a given height (to which it is raised by a pneumatic bulb), and the height of its rebound is taken as a measure of the hardness of the specimen. The type of instrument in most extensive use has a vertical scale graduated into 120 equal divisions, from which the scleroscope hardness number can be read off directly. The instrument is also made in another form, which indicates the hardness on a dial. To obtain accurate results, the hammer of the scleroscope must drop squarely onto a level polished surface. There is no direct proportionality between Brinell or Rockwell and scleroscope hardness numbers, but the relationship between these numbers for materials of different characteristics has been given considerable study, and curves showing this relationship have been published.

HEAT TREATMENT TERMINOLOGY

Heat treatment is an operation or series of operations involving the heating and cooling of metals or alloys in the solid state, for the purpose of obtaining certain desirable properties. Heating or cooling for the sole purpose of mechanical working is not covered by the term.

Quenching is immersing for the purpose of cooling, in liquids, gases, or solids.

Malleablizing is an annealing operation involving slow cooling, whereby combined carbon in white cast iron is converted into temper carbon, and in some cases is entirely removed from the iron. Temper carbon is free carbon in the form of rounded nodules made up of an aggregate of minute crystals.

Graphitizing is annealing of cast iron in such a way that some or all of the combined carbon is converted into uncombined carbon.

Carburizing or cementing consists in adding carbon to iron-base alloys by heating the metal below its melting point in contact with carbonaceous material.

Case-hardening consists in carburizing and subsequent hardening by suitable heat-treatment of all or part of the surface of a part of iron-base alloy.

Case is that portion of a carburized iron-base alloy part in which the carbon content has been substantially increased.

Core is that portion of a carburized irno-base alloy part in which the carbon content has not been substantially increased.

Cyaniding is a process of surface-hardening an iron-base alloy part by heating it to a suitable temperature and bringing it in contact with a cyanide salt, followed by quenching.

Normalizing consists in heating a part of iron-base alloy to above its upper critical range and allowing it to cool freely in the air.

FATIGUE OF METALS

It was observed by Wöhler about a century ago that if machine parts are subjected to repetitions of stress, they may fail eventually even though the maximum stress is well below the elastic limit. This led to the coinage of the word "fatigue of metals." A great deal of research work has been devoted to this subject since 1920, in this country particularly at the Engineering Experiment Station of the University of Illinois, under the direction of Professor H. F. Moore. The most interesting case is that in which a part is carried through cycles of stress between equal positive and negative limits, that is, where it is subjected to complete reversals of stress, as this applies to many machine parts.

Endurance Limit.—As regards wrought ferrous metals, Professor Moore's experiments showed that there is an endurance limit, that is, a unit stress below which a metal is capable of withstanding an indefinitely large number of reversals of stress. This endurance limit may be determined with a fair degree of accuracy by a short-time test in which the rise in temperature under reversed stress is measured.

Recently it has been found by Japanese metallurgists that the endurance limit may be determined in a short-time test with quite a fair degree of accuracy by measuring the electrical resistance of the specimen while subjected to alternating stresses. The specimens used in endurance tests consist of round bars of definite diameter and length, which are clamped in the head or heads of the testing machines, these heads being revolved at high speed usually by an electric motor. The specimen is subjected to a bending stress by suspending one or two weights from it, through the intermediary of annular ball bearings. One type of endurance testing machine is known as a cantilever type; in this the specimen overhangs the bearings and supports a weight at its free end. As the specimen revolves with the head of the machine, the direction of the stress on its fibres constantly changes, and there is a complete reversal of stress during the period of a half-revolution. In another type of machine the specimen is held at both ends and load is applied to it by means of two weights suspended from it at intermediate points through ball bearings. With this arrangement it is possible to so arrange things that the unit stress is constant over a certain length at the middle of the specimen.

Electric Short-Time Test.—In determining the endurance limit by means of electrical measurements, current is led through a part of the specimen in which the stress is constant at all sections. The current is led into and taken out of the revolving specimen by means of brushes, and the section of the specimen between brushes is included in one arm of a Wheatstone bridge, so that its electrical resistance can be measured.

If the machine is set in motion and the load on the specimen is gradually increased, it is found that up to a certain value of the unit stress, there is no increase in the electrical resistance of the specimen over that measured when it is stationary. From this point on the resistance increases quite rapidly, so that the point of change in resistance, which marks the endurance limit, is quite distinct. The explanation of the phenomenon is that at stresses below the endurance limit the deformation of the specimen is entirely elastic, but at higher stresses there is movement at the cleavage planes of the crystals of the metal, and this results in an increase in the resistance to the flow

of electric current. Thus the endurance limit of any particular metal can be very quickly determined. The same principle evidently underlies the short-time determination of the endurance limit by observing the temperature rise of the specimen, for when the stress exceeds the endurance limit and there is slippage at the cleavage planes of the crystals, this slippage naturally results in the generation of heat.

Relation to Other Properties.—The endurance limit is correlated with the ultimate tensile strength and the Brinell hardness number, and in a much smaller degree with the yield point and the elastic limit. No correlation has been found between the endurance limit and the ductility, the results of the Charpy impact test on notched bars, and repeated impact tests. It was found that the endurance limit increased slightly with the speed of stress reversal above 5000 r.p.m.

The results of tests on specimens drawn at various temperatures after oil quenching indicate that for the carbon steel specimens tested neither the ultimate tensile, the endurance limit for flexure, nor the ductility was appreciably affected by draws at a temperature lower than 600° Fahr. With nickel steel specimens for draws up to 400° Fahr. the ultimate tensile strength and the endurance limit for flexure diminished slightly, while the ductility increased a little. For draws at higher temperatures the changes in values of strength and ductility were more marked, the ultimate tensile strength and endurance limit increasing. Whatever advantages may be gained by draws below about 600° F., such as relief from internal stress and increased machineability, may be attained with little, if any, sacrifice of tensile strength or of endurance strength under flexure.

The results of tests on specimens subjected to severe tensile overstress before being tested in reversed bending indicated that a few aplications of stress well above the proportional elastic limit of the metal, lowers the endurance limit under subsequent reversed stress, the endurance being diminished by 22.9 per cent of its original value for one set of specimens. Placing the specimens in boiling water did not seem to have any effect on the endurance limit.

Failures of Keyed Shafts.—In shafts provided with keys, if torque is transmitted through the key, the crack starts at the junction of the side and the bottom of the keyway, and fracture finally occurs in the circumferential direction. This also applies to splined shafts. Under repeated horizontal stresses shaft fracture occurs in the direction either of the plane of principal stress or of maximum shear stress. The spiral fracture has been considered characteristic of hard steels, the circumferential of soft steel, but according to tests made by the Aero-Dynamics Department of the National Physical Laboratory the axial-circumferential fracture is normal for sound material, while a small flaw or a small hole drilled into the specimen increases the principal stress and leads to helicoidal fracture.

Summary of Professor Moore's Work.—Following is a summary of the results obtained in work on the fatigue of metals in the Engineering Experiment Station of the University of Illinois and reported in a Bulletin issued by the University in 1924.

(1) Additional evidence of the existence of an endurance limit for wrought ferrous metals is furnished by the results of re-tests of specimens which have withstood millions of cycles of stress without failure. Such re-tests not only failed to give evidence of damage to the metal by the millions of cycles of understress, but actually showed an increase of resistance to further repeated stress. This strengthening effect was most marked for those steels which are susceptible of improvement in static strength by cold working.

(2) Repeated stress at or below the original endurance limit of a wrought ferrous metal raises the endurance limit for that metal. A few cycles of repeated stress above the original endurance limit of a wrought ferrous metal lowers the endurance limit for that metal. A wrought ferrous metal injured by such overstress may have its strength partly, but rarely wholly, restored by polishing its surface or by repeated stress below the endurance limit. The static strength of a wrought ferrous metal is increased in the same general way as is the endurance limit, by repeated stress below the endurance limit.

(3) Under cycles of alternate axial tension and axial compression (completely reversed axial stress) specimens of wrought ferrous metals gave endurance limits averaging 64 per cent of the endurance limits for the same metals under cycles of reversed flexual stress. Probably the lower value found for the endurance limit under reversed axial stress is partly due to uneven stress-distribution in the short axial-stress specimens used. However, the stress-distribution in these specimens would in all probability be as regular as the stress-distribution in machine parts subjected to nominal axial stress. It is, accordingly, tentatively recommended that the endurance limit of wrought ferrous metals under reversed axial stress be considered as 60 per cent of the endurance limit under reversed flexural stress.

Tensile Strength and Endurance Limit.—(4) The static tensile strength and the endurance limit of wrought ferrous metals were found to be lower for specimens tested across the direction of rolling than for specimens tested along the direction of rolling. Test results for a number of specimens of wrought iron and of 0.37 per cent carbon steel indicate that the effect of the direction of rolling on the endurance limit is fully as great as the effect on the static ultimate tensile strength.

(5) Test results for 48 specimens of 0.37 per cent carbon steel cut from a 4-in. billet and subsequently heat-treated indicated some relative improvement of the strength of specimens tested across the direction of rolling as compared with specimens tested along the direction of rolling. The use of heat treating to improve the strength of steel across the direction of rolling is a problem worthy of further study.

(6) The endurance limit and the static strength of very low-carbon steel (ingot iron, 0.02 per cent carbon) were raised appreciably by heat treatment (a water quench) although the effect was much less marked than in the case of high-carbon steels and of alloy steels.

(7) For wrought ferrous metals the endurance limit for cycles of reversed shearing stress is about 53 per cent of that for cycles of reversed flexures. The endurance limits for reversed shearing stress were determined by tests of specimens under cycles of reversed torsion.

(8) Preliminary tests on specimens of annealed copper, annealed brass, annealed bronze and hot-rolled monel metal have not yet determined that these metals have an endurance limit. If such a limit exists it may be necessary to carry fatigue tests to hundreds of millions of cycles of stress to determine it.

Equations for Stresses.—(9) For flexual stress or for axial tensile stress either one of the following equations gives safe values of endurance limit for wrought ferrous metals under cycles of partially reversed stress or of stress varying from zero to a maximum:

$$S_r = S_{-2}\frac{r+3}{2} \text{ or } S_r = S_{-1}\sqrt{\frac{2}{1+r^2}}$$

in which S_r is the tensile unit-stress at the endurance limit for the range of stress represented by r; S_{-1} is the tensile strength unit-stress at the endurance limit for completely reversed stress; and r is the ratio of minimum stress to maximum stress for a cycle of stress. Data are lacking for determining the endurance limit for cases in which r is positive.

(10) For shearing stress the best data available indicate that for the wrought ferrous metals tested the endurance limit for cycles of stress varying from zero to a maximum is nearly twice the endurance limit for cycles of completely reversed stress. In the absence of more complete data it may be noted that equations corresponding to those given in the preceding paragraph give results which are on the safe side for repeated shearing stress.

Short-Time Thermal Test.—No direct experimental evidence is available in support of the following conclusion, but it seems to be in general accord with observed phenomena:

(11) It seems probable that the short-time test for determining endurance limit by the rise of temperature of a specimen subjected to some hundreds of cycles of reversed stress and the short-time test for determining endurance limit from the load-deflection diagram for a rotating beam determine the beginning of appreciable slip within the crystalline grains of the metal rather than the beginning of fatigue fracture. For wrought ferrous metals, however, with a Brinell hardness number less than about 375, this slip takes place under stresses at or slightly below the endurance limit, and these short-time tests seem safe tests for determining the endurance limits for such wrought ferrous metals if they are reasonably free from inclusions and flaws. In the absence of conclusive test data these short-time tests should not be regarded as safe tests for determining the endurance limit of cast metal or of non-ferrous metals.

Extension and Modification of the Theory.—The following three paragraphs summarize the statement of a theory of the fatigue of metals which may be regarded as an extension and modification of the well-known theory that fatigue failures in metals start from slip planes within the crystalline grains:

(12) Fatigue phenomena of metals are discussed on the theory that fatigue failure starts at some minute region of high localized stress and that its beginning is a tearing spot of minute pieces of metal. Fatigue failure proceeds by the spreading of the minute fissures thus formed. These fissures may be, but are not necessarily, the sequel of slip along cleavage planes within the crystalline grains of the metal.

(13) There is presented the further concept that repeated stress produces in a metal both strengthening effects and destructive effects. The strengthening effects are held to include the improvement of bearing between adjacent crystalline grains or other small particles of metal. The destructive effects are held to consist in the actual tearing apart of particles of the metal with the consequent formation of minute cracks, which have a tendency to spread.

Effects of Repeated Stresses.—(14) There is presented the further concept that for wrought ferrous metals the strengthening effects overbalance the destructive effects for cycles of stress below a fairly well-defined limiting stress, but that above this limiting stress the destructive effects predominate and failure occurs if the stress be repeated a sufficient number of times. This limiting stress is the endurance limit of the metal. The following conclusion is based on direct experimental evidence (microscopic examination of metal structure) and on a general study of the phenomena of mechanical tests of metals. It states a fact long since familiar to the careful student of the mechanics of materials but often overlooked by the casual student.

(15) The ordinary formulas of mechanics of materials are based on assumptions of homogeneity of material and of regularity of stress-distribution throughout the material, and these assumptions imply a much simpler condition than actually exists in metals and other materials of construction. However, these assumptions give a picture of stress-distribution which may be regarded as accurate in a general, statistical way. Even in the case of progressive failure under repeated stress the ordinary assumptions and formulas of mechanics of materials are found useful.

FATIGUE AND ELASTIC PROPERTIES OF SPRING STEEL

Research work on spring steels has been in progress in England since shortly after the war, and the results of one phase of this investigation were reported in a paper on "The Mechanical and Metallurgical Properties of Spring Steels as revealed by Laboratory Tests" presented to the Iron and Steel Institute (Great Britain) in 1929 by Dr. G. A. Hankins and Miss G. W. Ford. The tests were made on low chromium steel (0.56 per cent chromium), a high chromium steel (1.14 per cent chromium), a nickel-chromium steel (3.42 per cent nickel, 0.60 per cent chromium) and two water-quenched carbon steels (0.60 per cent and 0.46 per cent carbon, respectively).

The results obtained represent the properties of the materials as revealed by laboratory tests on prepared polished specimens, and the authors pointed out that the fatigue limits ascertained were much higher than would ordinarily be obtained from spring plates in the normal condition in which they are used in practice. Thirty-three fatigue-limit determinations were made on the steels, and the ratio of the fatigue limit in lbs. p. sq. in. to the Brinell number varied within the comparatively narrow range of 204 to 258, with a total mean value of 235. It appeared that the endurance limit (rotating-cantilever method) of a spring steel possessing a Brinell number between 320 and 500 and a martensitic structure, can be estimated roughly from the approximate rule that the fatigue limit in long tons per square inch is one-tenth of the standard Brinell number.

A typical nickel-chromium steel, which is not usually regarded as a spring steel, gave fatigue values which were very similar to those obtained from spring steels. It does not follow that the approximate equality of several of the steels in resistance to fatigue would also hold for plates or wires in which unknown disturbing surface factors are present. These fatigue determinations formed an essential preliminary, however, to a detailed examination of surface defects.

Results of determinations of the moduli of elasticity were fairly consistent for each steel, irrespective of the heat treatment, and, except for the low-chromium steel, the mean values for all the steels were about the same. Excluding the low-chromium steel, 62 determinations of the tensile modulus of elasticity were made, and the results varied from 29,500,000 to 30,900,000 lbs. p. sq. in. with a total mean value of 30,100,000 lbs. p. sq. in. Similarly, the modulus of rigidity (coefficient of elasticity in shear) varied from 11,300,000 to 11,900,000 lbs. p. sq. in. with a total mean value of 11,600,000 lbs. p. sq. in. For normal design purposes it was suggested that the values 30,100,000 and 11,600,000 lbs. p. sq. in. for the two elastic constants might be used with safety for all spring steels, irrespective of composition or heat treatment.

NORMAL AND ABNORMAL CARBURIZING STEELS

Carburized and case-hardened parts often give trouble from soft spots, and in the past such trouble has been a source of much mystification, since the cause cannot be traced by chemical analysis. Of two steels having exactly the same composition so far as the elements generally determined are concerned, one may give a uniform and entirely satisfactory case while the other may develop soft spots. It has been possible to distinguish between two types of case-hardening steel; those which give uniformly good results are known as normal steels, while the others are referred to as abnormal steels. The subject has been investigated chiefly by H. W. McQuaid and N. W. Ehn, metallurgists of the Timken Roller Bearing Co.

It was found by these investigators that the condition of the steel as it comes from the mills has an important influence on the structure of the case both before and after quenching, and that for good results in case-hardening the steel must contain a minimum of non-metallic impurities. The presence of dissolved oxides in the steel is particularly harmful.

From a lengthy series of tests relating to this subject and reported in a paper presented by Mr. Ehn to the American Society for Steel Treating in 1922, the following conclusions (in substance) were drawn:

In case-hardening steel parts, very much depends upon the qualities of the steel, and since these qualities cannot be determined by the usual physical and chemical tests, it is necessary to make a carburizing test. This test consists of a microscopical examination of carburized sections, which examination, if properly conducted, involves no large amount of work. After a little experience it becomes a very easy matter to determine from the carburized structure whether the steel is suitable for carburizing or not. Carburizing on a laboratory scale can be carried out in a few hours, especially if some kind of tight-fitting containers are employed. It must be understood that a test of one specimen from, for instance, a large heat of open-hearth steel is not a perfect safeguard against improper steel, as different parts of a heat often show different carburizing qualities. However, even one sample from each heat is likely to catch the worst cases, and if several samples from each heat are tested, a reasonable security against improper steel should be obtained. Tests of this kind constitute, moreover, a distinct warning to the steel mills not to send their off-heats to consumers who are known to submit the steel to tests of this kind.

Chromium a Safeguard.—For most carburizing purposes, a clean, well made, straight low-carbon steel (S.A.E. 1020) would be satisfactory, especially if hardened by spray quench to counteract small variations in the steel. Owing to the difficulty of obtaining a steel of this kind of uniform and proper quality in the open market, it may be advisable to specify a steel with from 0.30 to 0.50 per cent chromium for general carburizing purposes.

For parts subjected to a very severe service, steels with higher alloy contents, such as chromium, nickel, and vanadium, can be recommended, although for ordinary purposes they are out of the question, owing to the higher price. A slight percentage of chromium, such as suggested, would give a steel of good carburizing properties. As it is very unusual to find a chromium steel with abnormal properties, the use of such steel would not increase the cost very much; it would

improve rather than impair the machining qualities (as straight S.A.E. 1020 steel often tears under the tools on account of being too soft), would increase the rate of penetration in carburizing somewhat, would be beneficial for the hardening, and, last but not least, would necessitate carefully deoxidized heats, as otherwise so much chromium would be lost in the furnace that the chromium content in the finished steel would be lower than specified.

Oxides Objectionable Also in Heat-Treated Steels.—The influence of oxides in steel referred to in the foregoing is not restricted to carburized low-carbon steel. They will, if present, materially influence the results obtained in heat treating and hardening of any steel. Grain size and structure are influenced by the presence of oxides, if the steel is subjected to tough annealing operations, and so is the spheroidizing of the cementite in annealing high-carbon steel. It is an interesting fact that by carburizing high-carbon steels so as to obtain a carbon content of 1.10 per cent or over, a very sensitive test on the degree of deoxidation is obtained, and that in several instances it has been possible to prove by this method that failures in hardening were due to presence of oxides in the steel. A well-deoxidized, high-carbon steel should give a normal structure which is described by Mr. Ehn as follows:

The hyper-eutectoid zone consists of pearlite and free cementite sharply defined at the grain boundaries or lying as needles throughout the mass of the crystals. The grains are generally large and sharp-cornered in outline.

In very abnormal steel the hypereutectoid zone consists of pearlite, cementite and free ferrite. The cementite lies as small curly fragmnts in a mass of ferrite formed by the more or less complete divorce of the pearlite. The outlines of crystals are not well defined; they do not follow straight lines, and the crystal size is small as compared with the normal steel. In order to develop this structure clearly it is necessary that excess cementite be present—the more the better—and the most pronounced types of this structure therefore occur in the corners of the specimens, where an almost complete divorce of the pearlite may be obtained.

Investigations on normal and abnormal steels were conducted also at the Bureau of Standards, and the results of McQuaid and Ehn were largely confirmed there. It was found, however, that aside from its tendency to develop soft spots, the abnormal steel may have certain advantages over the normal type. It was shown that by somewhat more drastic quenching it is possible to harden the abnormal steel satisfactorily. Incidentally it was discovered that air dissolved in the quenching water has a detrimental effect.

Normality and abnormality were found to have their origin in the deoxidation treatment during the refining process. Various ferro-alloys are added to produce a deoxidizing influence on the steel, and it has been found that ferro-manganese, ferro-silicon and ferro-carbo-titanium give a normal hypereutectoid zone after carburizing, while aluminum and ferro-titanium free from carbon, when used in the deoxidizing process, produce an abnormal hypereutectoid zone.

GRAPHICAL SOLUTION OF HELICAL GEAR PROBLEMS

Helical gears are used in motor vehicles for speedometer drives, oil pump and distributor drives and occasionally for other purposes. In many cases the gear ratio is fixed and the center distance is approximately fixed. There then remain to be determined the numbers of teeth, the pitch and the helix angles of the driving and driven members respectively.

There are three general arrangements of helical gears, viz., with parallel shafts, with shafts at right angles, and with shafts at oblique angles. Problems in connection with helical gears mounted on parallel shafts are quite simple and easily solved. Helical gears mounted on shafts making an oblique angle with each other are used only very rarely. A simple graphical method for solving problems in connection

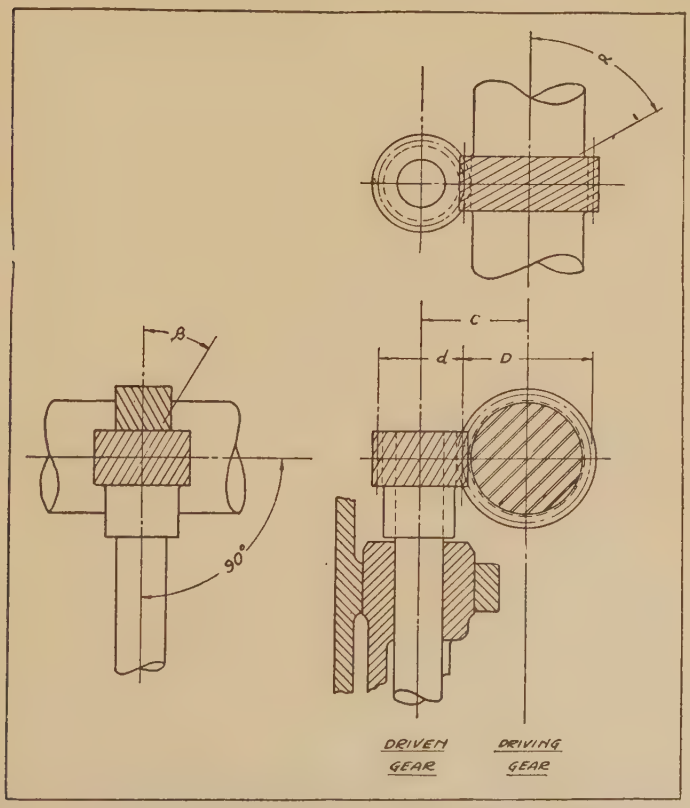


FIG. 1.—TYPICAL LAY-OUT OF HELICAL GEARS WITH SHAFTS AT 90°.

with gears having axes at right angles has been worked out by Robert P. Lewis, experimental design engineer of the Marmon Motor Car Co., and was published by him in Automotive Industries of July 28, 1928. This method may be explained with the aid of the accompanying illustrations, Figs. 1 to 3.

Figure 1 shows a set of helical gears with shafts at 90°. It is usually desirable that the center distance C be as short as possible in order to maintain a compact installation, thereby saving weight, material and cost. To obtain this ideal condition, it is necessary to design the gears about known centers, or to centers very closely approximating a given distance, so the following graphical analysis is based upon the assumption that the centers are known. This will also prove valuable in designing gears to be used on shafts for which the bearings have been located already.

As the minimum center distance for small power drives is usually determined by the shaft, hub and bearing diameters, clearances, etc., a preliminary layout should be made, similar to that shown in Fig. 1. In determining the centers for small power drives, the strength of the gears can be considered of secondary importance only, but in determining the centers for larger drives, the strength should be considered of greater importance, as by increasing the center distance the tooth pressure and the wear can be reduced.

Having obtained the center distance C by the preliminary layout, we are ready to proceed with the graph. The designer should remember that the larger the scale used, the more accurate will be the results. It is a good plan to make a few rough graphs full size, or twice size, until approximately the desired conditions are determined, and then make a more accurate graph to a larger scale.

Referring to Fig. 2, lay out lines EF and EG, 90° to each other. Select the normal diametral pitch P that is to be used, and assume the number of teeth N for the driving gear and the number n for the driven gear, keeping the desired ratio of $N:n$. Calculate the value of N/P and n/P, then locate the point O according to the scale selected, drawing horizontal and vertical lines OX and OY. Remember that all measurements must be made to the same scale. Now lay a scale, representing twice the center distance, $2C$, on the graph, the edge of the scale passing through point O and the ends just touching the lines EF and EG. The measurements D and d will be the pitch diameters, and the angles α and β the tooth helix angles of the driving and driven gears respectively.

Owing to the necessity of maintaining certain hub diameters, clearance limits and cutting possibilities, the pitch diameters of the gears must be held within certain maximum and minimum limits. A convenient method of doing this, by making a scale from a sheet of paper, is shown in Fig. 3. On the edge mark off twice the center distance ($2C$) to the scale that the graph is made to; also mark the maximum and minimum pitch diameters permissible. These can be determined by referring to the preliminary layout. Use this improvised scale in place of the standard scale. If the dimensions D and d on the graph fall within the limits shown on the scale, the design is practical. If not, the location of point O must be changed. To do this, change either the diametral pitch P or the number of teeth N and n, or both. If the number of teeth are changed, be sure that the desired ratio of $N:n$ is maintained. For some drives it is not necessary to maintain an exact ratio, an approximate ratio being satisfactory. This permits a more rapid solution, as fewer trials will be required.

The helix angles must also be considered, whether or not they are of a practical nature. The angles to be used should be chosen the same as when using the mathematical method. It is good practice to have the maximum thrust taken by the gear having the larger shaft and thrust bearing surface. This is usually the driving gear. The writer believes that for most cases, a good practical angle for the driving gear α would be between 55° and 60°, and for the driven gear β, between 30° and 35°. A few additional notes, for choosing the correct angle, are given at the end of this article. In some cases, such as speedometer drives, it is necessary to use extreme angles, these being as large as 70° to 85° for the driving gear α and as small as 5° to 20° for the driven gear β. Where the power required to be transmitted is very small, these extreme angles can be used satisfac-

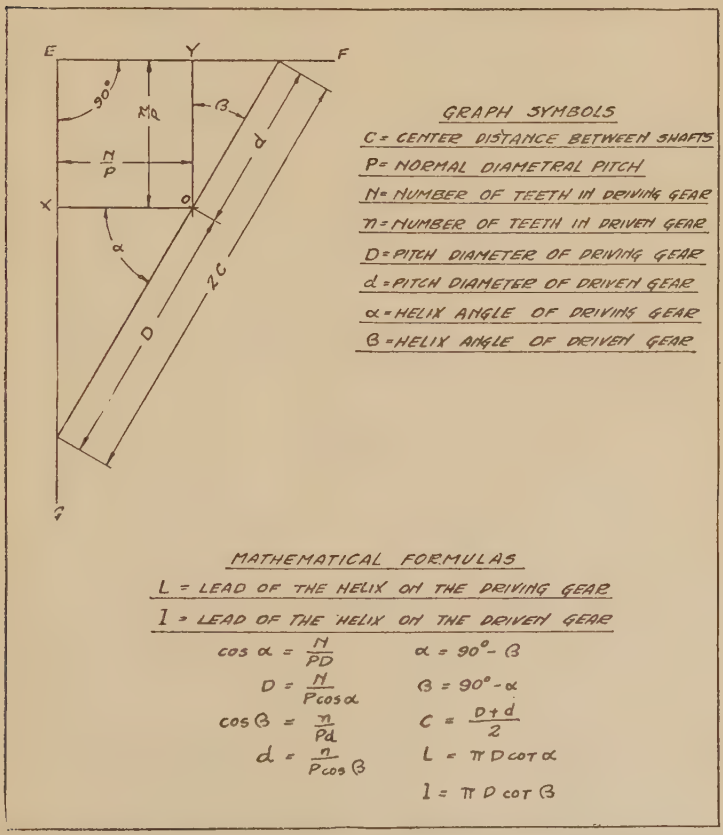

E
Y
F
90°
N/P
β
d
X
O
α
D
Z
GRAPH SYMBOLS
C = CENTER DISTANCE BETWEEN SHAFTS
P = NORMAL DIAMETRAL PITCH
N = NUMBER OF TEETH IN DRIVING GEAR
n = NUMBER OF TEETH IN DRIVEN GEAR
D = PITCH DIAMETER OF DRIVING GEAR
d = PITCH DIAMETER OF DRIVEN GEAR
α = HELIX ANGLE OF DRIVING GEAR
β = HELIX ANGLE OF DRIVEN GEAR
MATHEMATICAL FORMULAS
L = LEAD OF THE HELIX ON THE DRIVING GEAR
l = LEAD OF THE HELIX ON THE DRIVEN GEAR
cos α = N/PD
α = 90° − β
D = N/P cos α
β = 90° − α
cos β = n/Pd
C = (D + d)/2
d = n/P cos β
L = π D cot α
l = π D cot β

FIG. 2.—GRAPH FOR THE SOLUTION OF PROBLEMS ARISING IN THE DESIGN OF HELICAL GEARS WITH AXES AT RIGHT ANGLES.

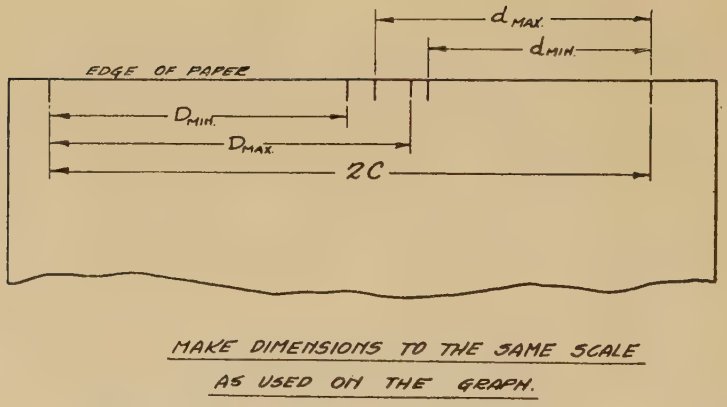


FIG. 3.—DRAWING FOR A SCALE TO BE MADE FROM PAPER, TO BE USED IN MAKING THE GRAPH FIG. 2.

torily. The angle α of the driving gear should never be less than 45° and that β, of the driven gear never more than 45°. If the angles shown on the graph do not appear practical, they should be changed by relocating the point O, this being done the same way as mentioned previously in connection with pitch diameters. Several trials may be necessary before all conditions are satisfied.

To avoid any small errors, the graphical measurements should be checked by the simple mathematical formulas given in Fig. 2.

Measure the pitch diameter of the driving gear D on the graph and solve for the angle α by the formula

$$\cos \alpha = \frac{N}{PD}.$$

Then find the angle β by the formula

$$\beta = 90° - \alpha,$$

and solve for the pitch diameter d of the driven gear by the formula

$$d = \frac{n}{P \cos \beta}.$$

Now find the actual center distance C by the formula

$$C = \frac{D + d}{2}.$$

This usually comes within a few thousandths of the original center distance used on the graph and layout. The calculated centers should be substituted for the original, the difference between the two being so small that the original layout should not be affected. However, if the centers must be exactly as the original, a few trial calculations are necessary, changing the angles and pitch diameters slightly, to get the desired results.

In Fig. 2 the formula for the lead is given for reference, as this should be calculated and placed on the details of the gears, for the machinist's information.

The graphical analysis does not consider the direction of rotation, direction and magnitude of thrust, strength and wearing condition of the teeth. These must be considered, the same as when using the mathematical method. Rules governing these conditions may be found in almost any mechanical engineering handbook.

The design of helical gears is of a more or less "elastic" nature, and by making a few permissable changes, almost any desired results may be obtained. Any one of the factors involved, such as center distance, pitch diameters, numbers of teeth, diametral pitch and helix angle, may be retained, provided others can be changed. The graphical analysis clearly shows how this may be easily and quickly done, and permits the possibilities of design to be seen at a glance.

A few rules regarding the design of helical gears with shafts at 90° follow:

1. The angle β should never be greater than 45°.
2. The power required to drive helical gears increases when the angle β is reduced. However, the increase is very small for angles between 45° and 30°, hence it may be disregarded in the case of drives consuming only small amounts of power.
3. The thrust in the direction of the driving gear axis increases when the angle β is reduced.
4. The thrust in the direction of the driven gear axis decreases when the angle β is reduced.
5. The force tending to separate the gears increases when the angle is reduced.
6. Increasing the pitch diameters reduces the pressure and wear on the teeth, and it also reduces the force tending to separate the gears.

PASSENGER-CAR BALLOON-TIRE LOAD AND INFLATION TABLE

Minimum Inflation Pressure, Pound		28	30	32	34	36
Tire Size	Wheel Diameter	Maximum Load per Tire in Pounds				
4.40	18 and 19	610	660	710	760	810
4.50	18 and 19	650	700	750	800	850
4.75	18 and 19	700	750	800	850	900
5.00	18 and 19	745	800	855	910	965
5.25	18 and 19	815	870	925	980	1035
5.50	18 and 19	880	940	1000	1060	1120
6.00	18 and 19	1000	1075	1150	1225	1300
(6.20)	18 and 19	1075	1155	1240	1320	1400
(6.75)	18 and 19	1200	1300	1400	1500	1600

BRITISH REGULATIONS GOVERNING BUS DESIGN

Regulations concerning features in bus design which affect the safety, comfort and health of the riding public and at the same time provide assurance against excessive road wear were issued by the British Minister of Transport in 1927. These included a good many provisions that had been in force previously in London only, while others were entirely new.

Not more than two doors or openings for passengers may be provided and neither may be on the right-hand side of the vehicle, viz.: the side removed from the footpath. All seats must be completely accessible from the inside. The minimum height inside at the center line of the vehicle from the floor to the lower edge of any depending part of the ceiling must be 63 in. where not more than 20 seats are provided and 70 in. where that seating capacity is exceeded.

When longitudinal seats are fitted there must be at least 54 in. from the back-rest of one seat to the back-rest of the one facing it. A seat length of 16 in. must be allowed for each passenger on both transverse and longitudinal seats.

Transverse seats must have a clear space of at least 25 in. in front of the back of each one; where such seats face one another there must be 19 in. leg clearance between them. Any form of seating accomodation across or in gangways is prohibited, except in vehicles having a folding top (e.g., motor coaches). Gangway width must be at least 12 in. wide at seat level and 14 in. above.

Interiors must be adequately ventilated without opening the main windows. All vehicles, open or closed, must have means whereby passengers can signal to the driver or conductor. If no conductor is carried, a rear mirror must be provided for the driver.

The lowest step at entrance must be no more than 17 in. above the ground or less than 10 in. when the vehicle is empty. Doorways must be at least 18 in. wide, or give at least 18 in. clear opening.

If the body has a permanent top and the entrance is not at the rear end, an emergency door opening outwards must be provided behind and clear of the rear wheels; it must be easily accessible to the passengers and so fastened as to enable it to be opened from both inside and outside. If the door handles at ordinary exits are on top of or inside the doors they must have double locks or safety catches.

Chassis Features.—It is specified that the front and rear wheel tracks must be as nearly as possible the same and that in no event must the front wheel track be less than that of the rear wheels. The track must not be less than 72 per cent of the overall width of the vehicle (which is limited to a maximum of 90 in.) in the case of a double deck bus, or 69 per cent in a single decker or coach.

When the axle weight of any vehicle exceeds four long tons, the diameter of the wheels, unless they have pneumatic tires, may not be less than 1010 mm.

The rear springs must be attached to or bear upon the axle as near to the wheels as possible, and their width apart, outside to outside, must be at least 50 per cent of the overall width. The front springs must be spaced not less than 42 per cent of the overall width on double deck buses, or 37 per cent on single deckers. Transverse springs are prohibited. Left-hand drive is also not allowed. No hub cap may project more than 3½ in. beyond the rim of the wheel.

Location of Fuel Tanks.—Fuel tanks must be so placed that any overflow cannot fall upon woodwork, or accumulate where it can be readily ignited. The exhaust pipe must not be fixed in such a position that oil, or any vaporizable or inflammable materials, is likely to be dropped on to it. Carburetors must be so placed, incased or screened as to prevent any leakage of fuel falling on to the ignition apparatus or any wire or connections carrying electric current.

No passenger shall be permitted on the right-hand side of the driver, the rule of the road being "Keep to the Left." Where passengers are carried on the left of the driver a space of at least 18 in. from the center of the steering column to the left of it shall be reserved for the driver and divided off by a solid partition at least 9 in. high.

Vehicles not fitted with running boards, or continuous steps between front and rear wheels or fenders, must be fitted with guard rails inclosing the full length of the space between the wheels or fenders.

Dimensional and Weight Limits.—The overall length of buses and trucks weighing more than 6720 lb. (three long tons) is limited for the first time, and while four-wheeled vehicles are given a maximum of 27 ft. 6 in. (or 26 ft. if a trailer is drawn), vehicles of the rigid six-wheeled type can be of any length up to 30 ft.

In the case of four-wheeled buses the total laden axle weight must not be over 9 long tons and the weight of any axle must not exceed 5½ tons. Rigid six-wheeled buses can, however, have a total laden weight of 12 tons and six wheeled trucks 19 tons, though the maximum unladen weight for both buses and trucks with six wheels is 10 tons.

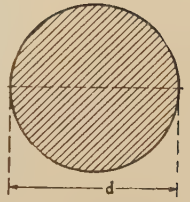
A limitation is set upon overhang behind the rear axle center of four-wheelers, and a line 4 in. behind a point midway between the two rear axles of six-wheelers. The maximum overhang beyond those points is 7-24ths of the overall length of the vehicle.

Then, too, a minimum ground clearance is specified; in the case of four-wheelers it is 10 in. for 14 ft. 6 in. from the front of the vehicle and in that of six-wheelers 10 in. for 13 ft. from the front.

The diameter of the turning circle is also covered; it may not be greater than 60 ft. where the overall length is 26 ft. or less, or 66 ft. where the overall length is more than 26 ft.

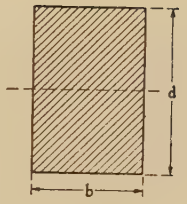
Brakes.—Every vehicle must have at least two independent sets of brakes, one if not both acting directly upon all the wheels except the front (though front brakes are not barred) and not through the transmission. The brakes will not be deemed independent if the failure of any part of the operating mechanism of one of them could by any chance affect the operation of the other. There is only one exception to the provision that all parts must be independent and that relates to the drums, which may be common to both sets. In the case of rigid six-wheelers, one set of brakes may act upon the middle axle and the other upon the rearmost axle.

In a circular sent by the Ministry of Transport to public authorities explaining the order it said among other things that it considered the higher maximum weight for rigid-frame six-wheelers justified on the ground that vehicles of this type, if suitably constructed, offer special advantages from the standpoint of road wear, not only by reason of the wider distribution of weight but also because of the reduction of impact between the wheels and inequalities of the road.



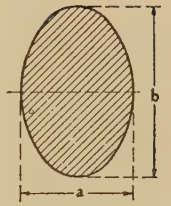
$$I = \frac{\pi d^4}{64}$$

$$Z = \frac{\pi d^3}{32}$$



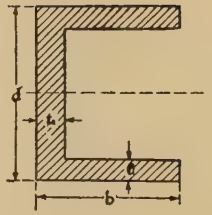
$$I = \frac{b d^3}{12}$$

$$Z = \frac{b d^2}{6}$$



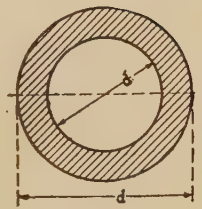
$$I = \frac{\pi a b^3}{64}$$

$$Z = \frac{\pi a b^2}{32}$$



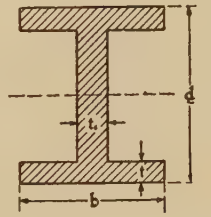
$$I = \frac{b d^3 - (b - t_1)(d - 2t)^3}{12}$$

$$Z = \frac{b d^3 - (b - t_1)(d - 2t)^3}{6 d}$$



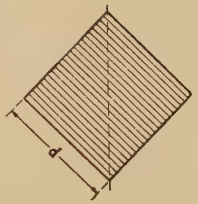
$$I = \frac{\pi(d^4 - d_1^4)}{64}$$

$$Z = \frac{\pi(d^4 - d_1^4)}{32\, d}$$



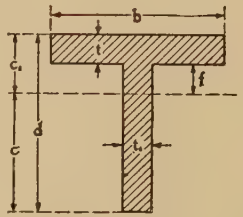
$$I = \frac{b d^3 - (b - t_1)(d - 2t)^3}{12}$$

$$Z = \frac{b d^3 - (b - t_1)(d - 2t)^3}{6 d}$$



$$I = \frac{d^4}{12}$$

$$Z = \frac{d^3}{6}$$



$$I = \frac{1}{3}\left[b(c_1^3 - f^3) + t_1(c^3 + f^3)\right]$$

$$Z = \frac{1}{6\, d}\left[b(c_1^3 - f^3) + t_1(c^3 + f^3)\right]$$

MOMENTS OF INERTIA AND SECTION MODULI OF COMMON SECTIONS.

National Coarse Thread Series

(Form of thread known as U. S. Standard)

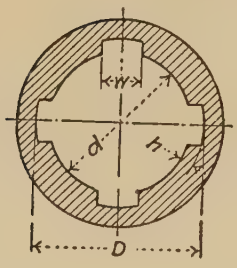
Size	Threads Per Inch	Outside (Major) Diam.	Depth of Thread	Tap Drill Size
		Inches	Inches	Inches
5	40	.125	.01624	.0995
6	32	.138	.02030	.1063
8	32	.164	.02030	.1324
10	24	.190	.02706	.1477
12	24	.216	.02706	.1732
1/4	20	.250	.03248	.1990
5/16	18	.3125	.03608	.2559
3/8	16	.375	.04059	.3110
7/16	14	.4375	.04639	.3642
1/2	13	.500	.04996	.4219
9/16	12	.5625	.05413	.4776
5/8	11	.625	.05905	.5315
3/4	10	.750	.06495	.6480
7/8	9	.875	.07217	.7615
1	8	1.000	.08119	.8723
1 1/8	7	1.125	.09279	.9789
1 1/4	7	1.250	.09279	1.1024
1 1/2	6	1.500	.10825	1.3281
1 3/4	5	1.750	.12990	1.5453
2	4 1/2	2.000	.14434	1.7717

National Fine Thread Series

(Form of thread known as U. S. Standard)

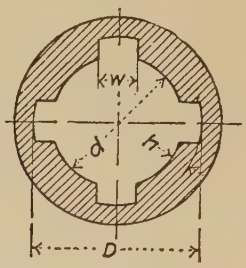
Size	Threads Per Inch	Outside (Major) Diam.	Depth of Thread	Tap Drill Size
		Inches	Inches	Inches
8	36	.164	.01804	.1360
10	32	.190	.02030	.1590
12	28	.216	.02320	.1800
1/4	28	.250	.02320	.2130
5/16	24	.3125	.02706	.2703
3/8	24	.375	.02706	.3320
7/16	20	.4375	.03248	.3860
1/2	20	.500	.03248	.4492
9/16	18	.5625	.03608	.5062
5/8	18	.625	.03608	.5687
3/4	16	.750	.04059	.6875
7/8	14	.875	.04639	.8024
1	14	1.000	.04639	.9274
1 1/8	12	1.125	.05413	1.0401
1 1/4	12	1.250	.05413	1.1651
1 1/2	12	1.500	.05413	1.4153

S. A. E. Standard Splined Joints.



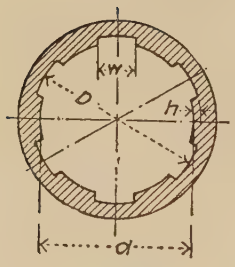
4A

w = 0.241D
h = 0.075D
d = 0.850D



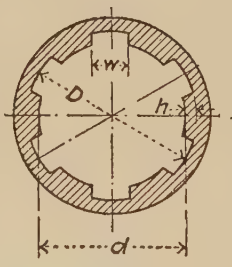
4B

w = 0.241D
h = 0.125D
d = 0.750D



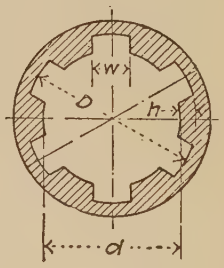
6A

w = 0.250D
h = 0.050D
d = 0.900D



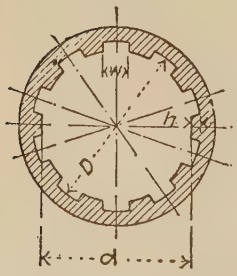
6B

w = 0.250D
h = 0.075D
d = 0.850D



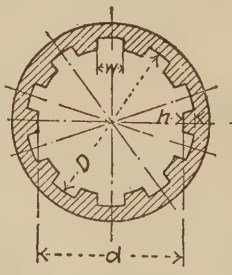
6C

w = 0.250D
h = 0.100D
d = 0.800D



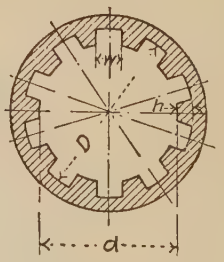
10A

w = 0.156D
h = 0.045D
d = 0.910D



10B

w = 0.156D
h = 0.070D
d = 0.860D

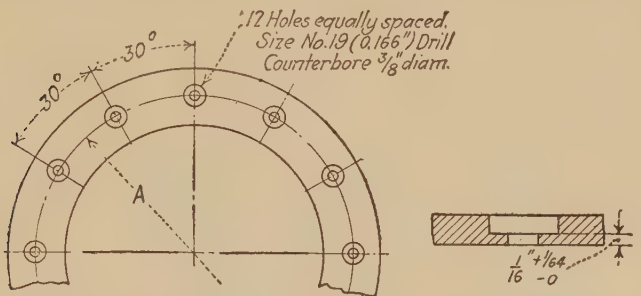


10C

w = 0.156D
h = 0.095D
d = 0.810D

DIMENSIONS FOR CLUTCH FACINGS

The inside and outside diameters of clutch discs of both woven and molded asbestos, and the thickness and tolerances thereon, as well as the number and location of rivet holes are covered by the S. A. E. Recommended Practice given below.



MULTIPLE-DISC CLUTCH-FACING SIZES

Woven Type		Molded Type		
Outside Diameter $+0.000,-\frac{1}{16}$	Inside Diameter $+\frac{1}{16},-0.000$	Outside Diameter $+0.000,-\frac{1}{32}$	Inside Diameter $+\frac{1}{32},-0.000$	A $\pm\frac{1}{32}$
$8\frac{1}{2}$	6	$8\frac{1}{2}$	6	$7\frac{3}{16}$
$8\frac{1}{2}$	$6\frac{1}{4}$	$8\frac{1}{2}$	$6\frac{1}{4}$	$7\frac{7}{16}$
$8\frac{1}{2}$	$6\frac{1}{2}$	$8\frac{1}{2}$	$6\frac{1}{2}$	$7\frac{7}{16}$
$8\frac{1}{4}$	6	$8\frac{1}{4}$	6	$7\frac{3}{16}$
$8\frac{1}{4}$	$6\frac{1}{4}$	$8\frac{1}{4}$	$6\frac{1}{4}$	$7\frac{3}{16}$
$8\frac{1}{4}$	$6\frac{1}{2}$	$8\frac{1}{4}$	$6\frac{1}{2}$	$7\frac{7}{16}$
8	$5\frac{3}{4}$	8	$5\frac{3}{4}$	$6\frac{15}{16}$
8	6	8	6	$6\frac{15}{16}$
8	$6\frac{1}{4}$	8	$6\frac{1}{4}$	$7\frac{3}{16}$

The thickness of both woven and molded types of facing shall be 5/32 ± 0.005 in. Limits do not apply to parallelism of the faces.

SINGLE-PLATE CLUTCH-FACING SIZES

Size	Woven Type		Molded Type	
	Outside Diameter $+0.000,-\frac{1}{16}$	Inside Diameter $+\frac{1}{16},-0.000$	Outside Diameter $+0.000,-\frac{1}{32}$	Inside Diameter $+\frac{1}{32},-0.000$
8	$7\frac{7}{8}$	$5\frac{3}{8}$	$7\frac{3}{4}$	$5\frac{1}{2}$
9	$8\frac{7}{8}$	$6\frac{1}{8}$	$8\frac{3}{4}$	$5\frac{3}{4}$
10	$9\frac{7}{8}$	$6\frac{3}{4}$	$9\frac{3}{4}$	$6\frac{1}{4}$
11	$10\frac{7}{8}$	$6\frac{3}{4}$	$10\frac{3}{4}$	$6\frac{1}{2}$
12	$11\frac{7}{8}$	$7\frac{1}{4}$	$11\frac{3}{4}$	$6\frac{1}{2}$

The thickness of the woven type of facing shall be 1/8 or 5/32 ± 0.005 in.

The thickness of the molded type of facing shall be 9/64 ± 0.005 in.

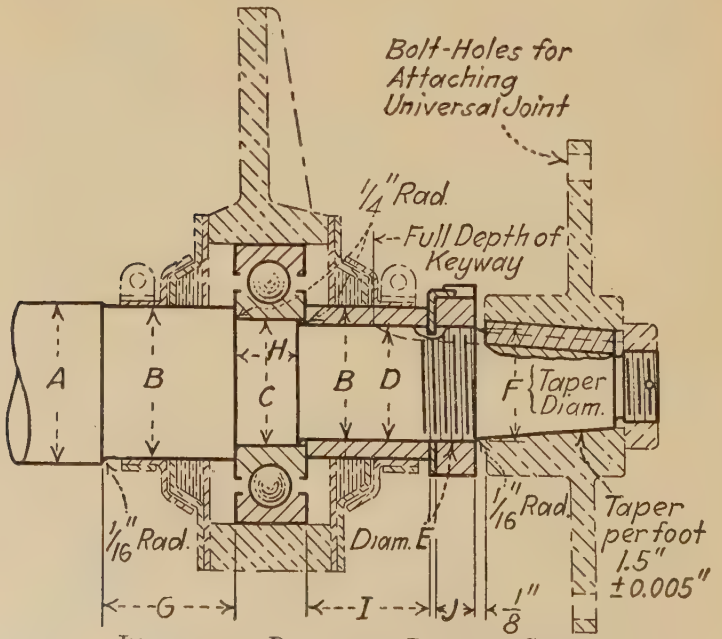
Limits do not apply to parallelism of the faces but limits for parallelism in present practice range from 0.002 to 0.005 in.

Wrought Iron Pipe and Pipe Threads.

(The thread is an American Standard)

Nominal Size	Inside Diameter Pipe	Outside Diameter Pipe	Threads per Inch	Length of Perfect Threads	Outside Diameter. Small End of Pipe	Tap Drill Size
1/8	0.270	0.405	27	0.190	0.393	21/64
1/4	0.364	0.540	18	0.290	0.522	7/16
3/8	0.494	0.675	18	0.300	0.656	9/16
1/2	0.623	0.840	14	0.390	0.816	45/64
3/4	0.824	1.050	14	0.400	1.025	29/32
1	1.048	1.315	11½	0.510	1.283	1 9/64
1¼	1.380	1.660	11½	0.540	1.627	1 31/64
1½	1.610	1.900	11½	0.550	1.866	1 23/32

Ordinary pipe threads taper ¾ in. to the foot. The thread form is that of a modified V. The same sizes are also made in the form of straight threads, and if straight threads are wanted they should be specified. The depth of the thread is equal to 0.8 number of threads per inch.



INTERMEDIATE BEARING FOR PROPELLER SHAFT.

Intermediate Bearing for Propeller Shaft

(See drawing on opposite page)

The type of housing shown is recommended as a satisfactory design, but is not a part of the standard. No slip-joint in the forward section of the shaft is required, as clearance is provided in the housing for the bearing and shaft to slide longitudinally. Where a slip-joint is used, the housing should be designed according to requirements. The shafts and bearings recommended are considered adequate for all usual installations. Where heavier duty may require larger bearings, the shaft dimensions should be determined by the manufacturer and the purchaser.

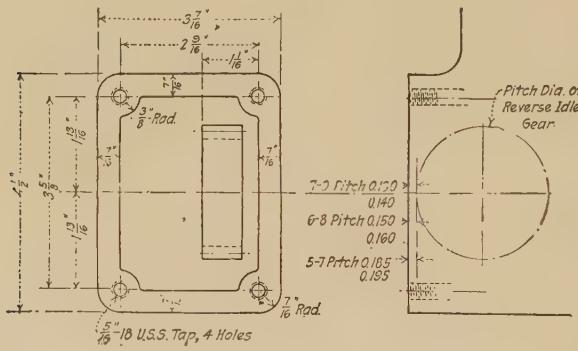
DIMENSIONS FOR THREE-JOINT PROPELLER-SHAFT MIDSHIP MOUNTING

Tube Diam., *A*, in.	2	2	2¼	2½	3
B, +0.005, −0 in.	1.6870	1.8750	2.1250	2.3750	2.5620
C, max.	1.3780	1.5748	1.7717	1.9685	2.1654
C, min.	1.3772	1.5740	1.7709	1.9677	2.1646
D, +0, −0.003 in.	1.3740	1.4990	1.6240	1.8740	2.1240
E	1⅜–18	1½–18	1⅝–16	1⅞–16	2⅛–16
Shaft end, *F*	1¼	1⅜	1½	1¾	2
G, in.	1⅝	1¾	1⅞	2	2⅛
H, in.	49/64	27/32	59/64	1	1 5/64
I, in.	1 9/16	1 11/16	1 13/16	1 15/16	2 1/16
J, in.	7/16	7/16	9/16	9/16	11/16
Bearing No.	307	308	309	310	311

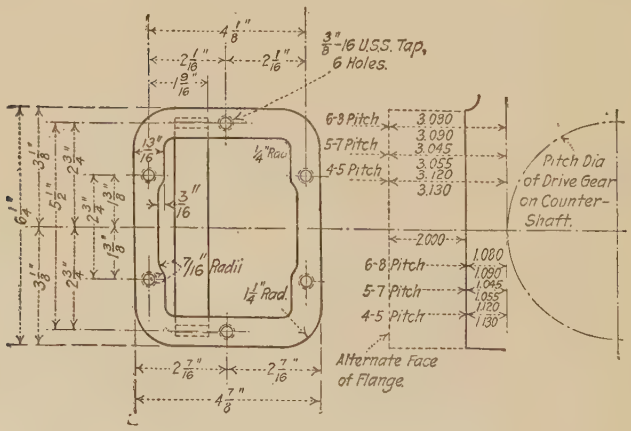
Note.—The oil-retainer rings should be adjusted for minimum clearance after the housing and shaft are assembled in the chassis.

POWER TAKE-OFF AND TIRE-PUMP MOUNTING

Drives for tire pumps or for auxiliary purposes (power take-offs) are often taken from the transmissions of motor trucks (tire-pump drives also from those of passenger cars) and the S. A. E. has standardized the dimensions of the opening in the transmission housing, the threaded holes and their locations, and the location of the gear from which the drive is taken, for both a small and a large power take-off, as shown by the drawings below.



SMALL MOUNTING



LARGE MOUNTING

If no power take-off or tire pump drive is required, the opening in the transmission housing is closed by a cover.

STANDARD ROLLER CHAINS, HEAVY SERIES

The table below gives the dimensions of parts and breaking strengths of the heavy series of roller chains as standardized by the S. A. E. These chains are suitable for rear end drives of slow and moderate-speed vehicles.

HEAVY SERIES ROLLER-CHAIN DIMENSIONS

Chain Number[1]	Pitch	Roller Diameter	Pin Diameter	Thickness of Inside Plates	Minimum Chain Width	Measuring Load, Lb.	Approx. Test Load, Lb	Minimum Breaking-Strength,[2] Lb.
30W	3/8	0.250	0.125	0.050	0.250	12	376	940
40W	1/2	0.3125	0.156	0.060	0.3125	23	740	1850
50W	5/8	0.400	0.200	0.080	0.375	44	1400	3500
60W	3/4	0.469	0.234	0.094	0.500	63	2018	5045
80W	1	0.625	0.3125	0.125	0.625	119	3820	9550
100W	1 1/4	0.750	0.375	0.156	0.750	176	5625	14063
120W	1 1/2	0.875	0.4375	0.1875	1.000	242	7760	19400
140W	1 3/4	1.000	0.500	0.220	1.125	319	10220	25550
160W	2	1.125	0.5625	0.250	1.250	406	13000	32520
200W	2 1/2	1.550	0.781	0.3125	1.5625	792	25300	63350
240W	3	1.900	0.9375	0.375	1.875	1145	36600	91585
320W	4	2.500	1.250	0.500	2.500	2042	65300	163360
400W	5	3.000	1.500	0.625	3.000	2944	94200	235550
	Formulas	Approx. $\frac{5}{8}$ P	Approx. $\frac{1}{2}$ roller diam.	$\frac{1}{8}$ P	Approx. 0.625 P	0.0125×Min. Breaking Strength	0.40×Min. Breaking Strength	$\frac{105000 \times \text{Pin Diam.}^2}{-700}$

[1] Chains are numbered in accordance with the following system—The left-hand figure denotes the number of one-eighth inches in the pitch. The final figure denotes the roller diameter as follows: *0*, the heavy series roller diameter; *1*, the medium series roller diameter, and *2*, the light series roller diameter. The letters *W* or *N* denote whether the chains are of the wide or narrow type. All S. A. E. Standard chains are of the wide type.

[2] The minimum breaking strengths specified are the loads at which the chains may break or become permanently stretched, being based on static tests. The actual *working* loads may range from one sixth to one fiftieth of the ultimate strength, depending upon the speed, the number of teeth, the conditions of lubrication and the pulsations occurring at either power or load end. Chains should not be selected on the basis of breaking strength.

Chain No. 50W shall be used for motorcycle driving chains.

BALL STUDS

The following table of dimensions for ball studs forms an S. A. E. Recommended Practice. These studs are used chiefly for steering connections.

Size	A		B	C	D	E	F	G	H	J		K	L	M	N	Threads Per Inch	O	Drill No. (P)
	Nom.	Tol. Plus or Minus								Nom.	Tol. Plus or Minus							
½	21/64	0.005	7/32	7/16	21/32	7/16	½	19/32	9/32	13/32	0.002	3/8	1/32	5/16	1/2	24	3/32	48
5/8	27/64	0.005	9/32	9/16	27/32	9/16	5/8	23/32	11/32	15/32	0.002	1/2	3/64	3/8	5/8	24	3/32	48
¾	33/64	0.005	5/16	11/16	1 5/32	11/16	¾	27/32	13/32	17/32	0.002	5/8	1/16	7/16	¾	24	3/32	36
7/8	39/64	0.005	3/8	3/4	1 5/16	3/4	7/8	2 1/32	15/32	5/8	0.002	3/4	1/16	1/2	7/8	20	1/16	36
1	31/32	0.010	7/16	13/16	1 13/32	13/16	1	2 5/32	17/32	21/32	0.002	1	1/16	9/16	1	20	1/16	28
1⅛	1 7/32	0.010	½	29/32	1 17/32	29/32	1⅛	2 11/32	19/32	3/4	0.002	1 1/8	3/32	5/8	1⅛	18	5/32	28
1¼	1 13/32	0.010	9/16	15/16	1 21/32	15/16	1 1/8	2 17/32	21/32	25/32	0.002	1 1/4	3/32	11/16	1¼	18	5/32	28
1⅜	1 19/32	0.010	5/8	1 1/8	1 25/32	1 1/16	1 1/4	2 23/32	23/32	7/8	0.002	1 3/8	1/8	3/4	1⅜	16	3/16	23
1½	1 25/32	0.010	11/16	1 1/4	1 29/32	1 3/16	1 3/8	2 31/32	25/32	1	0.002	1 1/2	1/8	13/16	1½	16	3/16	23
1¾	1 31/32	0.010	¾	1 3/8	2 1/32	1 5/16	1 7/16	3 3/32	27/32	1 1/32	0.002	1 3/4	1/8	7/8	1¾	14	7/32	28
2	2 3/32	0.015	13/16	1 1/2	2 1/8	1 7/16	1 1/2	3 5/32	1 1/8	1 1/8	0.003	2	1/8	15/16	2	14	7/32	28
2¼	2 9/32	0.015	1	1 11/16	2 1/4	1 11/16	1 5/8	3 7/16	1 1/4	1 1/4	0.003	2 1/4	1/8	1 1/16	2¼	12	1/4	11
2½	2 17/32	0.015	1 1/8	1 7/8	3	1 7/8	1 7/8	3 11/16	1 3/8	1 3/8	0.003	2 1/2	1/8	1 3/16	2½	12	1/4	11
3	2 25/32	0.015	1 1/4	2 1/4	3 1/2	2 1/4	2 1/8	4 5/16	1 5/8	1 5/8	0.003	3	1/8	1 7/16	3	12	1/4	2

BALL STUDS—*Continued*

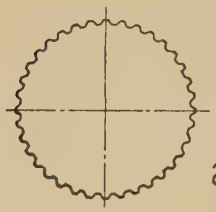
The table on this page forms part of the S. A. E. Recommended Practice on Ball Studs. It relates to the fastening of the nuts in steering levers or other parts.

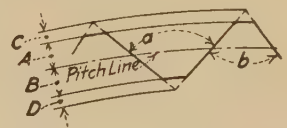
Size	Q		R	S	T		U		V	W		X	Y
	Nom.	Tol. Plus 0 Minus	max.		Nom.	Tol. Plus 0 Minus	Nom.	Tol. Plus 0 Minus		Nom.	Tol. Plus or Minus		
1/2	9/16	3/64	19/32	17/64	1/8	3/64	13/64	2/64	3/32	0.329	0.001	5/64	27/32
5/8	11/16	3/64	23/32	21/64	1/8	3/64	16/64	2/64	3/32	0.376	0.001	5/64	32/32
3/4	25/32	3/64	29/32	25/64	3/32	3/64	17/64	2/64	1/8	0.446	0.001	6/64	1 5/32
7/8	25/32	3/64	29/32	7/16	1/4	3/64	17/64	2/64	9/64	0.509	0.001	7/64	1 5/16
1	15/16	1/16	31/32	17/64	3/16	1/16	9/32	3/64	9/64	0.578	0.002	1/8	1 3/8
1 1/8	1 3/64	1/16	1 7/64	21/64	3/16	1/16	9/32	3/64	5/32	0.642	0.002	5/64	1 13/32
1 1/4	1 1/8	1/16	1 1/4	25/64	1/4	1/16	5/16	3/64	11/64	0.725	0.002	5/64	1 1/16
1 3/8	1 3/32	1/16	1 9/32	7/16	1/4	1/16	11/32	3/64	3/16	0.855	0.002	5/64	2
1 1/2	1 5/32	1/16	1 11/32	29/64	1/4	1/16	11/32	3/64	7/32	1.039	0.002	5/64	2 1/32
1 5/8	1 5/16	1/16	1 13/32	15/32	1/4	1/16	11/32	3/64	1/4				
1 3/4	1 13/32	1/16	1 15/32	31/64	1/4	1/16	11/32	3/64	1/4				
1 7/8	1 7/16	1/16	1 17/32	1/2	1/4	1/16	11/32	3/64	1/4				
2	1 15/32	1/16	1 19/32	33/64	1/4	1/16	11/32	3/64	1/4	1.147	0.002	5/64	2 5/64
2 1/4	1 23/32	1/16	1 27/32	37/64	1/4	1/16	11/32	3/64	1/4	1.304	0.002	5/64	2 13/64
2 1/2	2 1/32	1/16	2 1/32	41/64	1/4	1/16	11/32	3/64	1/4	1.468	0.002	5/64	2 29/64
3	2 7/32	1/16	2 7/32	49/64	1/4	1/16	11/32	3/64	1/4	1.733	0.002	5/64	2 45/64

SERRATED FITTINGS

The dimensions for serrated fittings given on this and the following page are an S. A. E. Recommended Practice. These fittings are much used for steering gear shafts, where it is desired to have a secure joint without reducing the strength of the shaft unnecessarily.

STRAIGHT TYPE



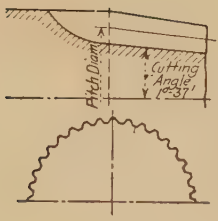


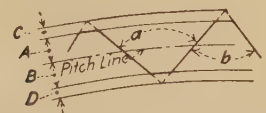
A and B are nominally equal
A + B = Depth of Cut
C in Hole and D in Shaft may be made as desired by each individual manufacturer beyond minimum and maximum as specified in table.
N = Number of Serrations

Nominal Diam.	Pitch Diam.		N	a, deg.	b, deg.	Hole			Shaft		
						Large Diam. Min.	Small Diam.		Outside Diam.		Inside Diam. Max.
	Max.	Min.					Max.	Min.	Max.	Min.	
1/8	0.122	0.120	36	90	80	0.125	0.118	0.117	0.124	0.123	0.116
3/16	0.182	0.180	36	90	80	0.187	0.176	0.175	0.186	0.185	0.174
1/4	0.243	0.241	36	90	80	0.250	0.235	0.234	0.249	0.248	0.233
5/16	0.303	0.301	36	90	80	0.312	0.293	0.292	0.311	0.310	0.291
3/8	0.363	0.361	36	90	80	0.375	0.352	0.351	0.374	0.373	0.350
1/2	0.485	0.483	36	90	80	0.500	0.469	0.468	0.499	0.498	0.467
5/8	0.605	0.603	36	90	80	0.625	0.584	0.583	0.624	0.623	0.582
3/4	0.733	0.731	48	90	82½	0.750	0.716	0.714	0.749	0.747	0.713
7/8	0.855	0.853	48	90	82½	0.875	0.835	0.833	0.874	0.872	0.832
1	0.977	0.975	48	90	82½	1.000	0.954	0.952	0.999	0.997	0.951
1 1/8	1.098	1.096	48	90	82½	1.125	1.071	1.069	1.124	1.122	1.068
1 1/4	1.220	1.218	48	90	82½	1.250	1.190	1.188	1.249	1.247	1.187
1 3/8	1.342	1.340	48	90	82½	1.375	1.309	1.307	1.374	1.372	1.306
1 1/2	1.464	1.462	48	90	82½	1.500	1.428	1.426	1.499	1.497	1.425
1 3/4	1.708	1.706	48	90	82½	1.750	1.666	1.664	1.749	1.747	1.663
2	1.952	1.949	48	90	82½	2.000	1.904	1.902	1.999	1.997	1.901
2 1/4	2.196	2.193	48	90	82½	2.250	2.142	2.140	2.249	2.247	2.139
2 1/2	2.440	2.437	48	90	82½	2.500	2.380	2.378	2.499	2.497	2.377
2 3/4	2.684	2.681	48	90	82½	2.750	2.618	2.616	2.749	2.747	2.615
3	2.928	2.925	48	90	82½	3.000	2.856	2.854	2.999	2.997	2.853

All dimensions in inches.

TAPER TYPE





A and B are nominally equal at Large End Only
A + B = Depth of Cut
C in Hole and D in Shaft may be made as desired by each individual manufacturer beyond minimum and maximum as specified in table
N = Number of Serrations
Length of Full Serration on Shaft = Normal Diam.

Nominal Diam.	Pitch Diam.		N	a, deg.	b, deg.	Hole			Shaft		
						Large Diam. Min.	Small Diam.		Outside Diam.		Inside Diam. Max.
	Max.	Min.					Max.	Min.	Max.	Min.	
½	0.485	0.483	36	90	80	0.500	0.469	0.468	0.499	0.498	0.467
⅝	0.605	0.603	36	90	80	0.625	0.584	0.583	0.624	0.623	0.582
¾	0.733	0.731	36	90	80	0.750	0.716	0.714	0.749	0.747	0.713
⅞	0.855	0.853	36	90	80	0.875	0.835	0.833	0.874	0.872	0.832
1	0.977	0.975	36	90	80	1.000	0.954	0.952	0.999	0.997	0.951
1⅛	1.098	1.096	36	90	80	1.125	1.071	1.069	1.124	1.122	1.068
1¼	1.220	1.218	36	90	80	1.250	1.190	1.188	1.249	1.247	1.187
1⅜	1.342	1.340	36	90	80	1.375	1.309	1.307	1.374	1.372	1.306
1½	1.464	1.462	36	90	80	1.500	1.428	1.426	1.499	1.497	1.425
1¾	1.708	1.706	36	90	80	1.750	1.666	1.664	1.749	1.747	1.663
2	1.952	1.949	48	90	82½	2.000	1.904	1.902	1.999	1.997	1.901
2¼	2.196	2.193	48	90	82½	2.250	2.142	2.140	2.249	2.247	2.139
2½	2.440	2.437	48	90	82½	2.500	2.380	2.373	2.499	2.497	2.377
2¾	2.684	2.681	48	90	82½	2.750	2.618	2.616	2.749	2.747	2.615
3	2.928	2.925	48	90	82½	3.000	2.856	2.854	2.999	2.997	2.853

All dimensions in inches and apply to large end of taper only.
Taper ¾ in. per ft. on outside diameter.
Cutting angle 1 deg. 37 min.

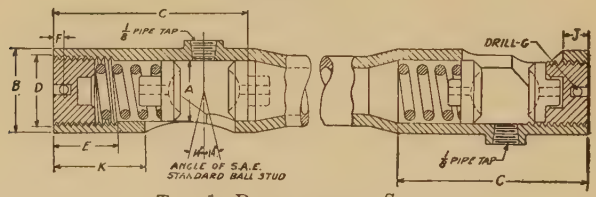
BRAKES FOR INDUSTRIAL TRUCKS AND TRACTORS

According to an S. A. E. Recommended Practice, brakes for industrial trucks and tractors should be so designed that they are released by depressing the brake pedal and applied by removing the foot. The method of applying the brake by the operator leaving the seat is discouraged.

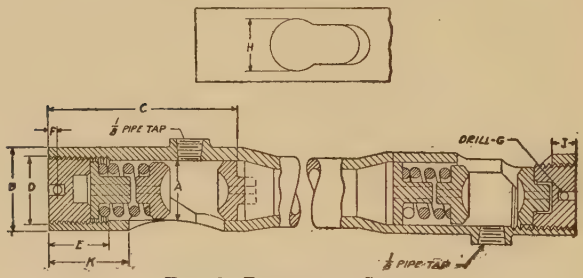
STEERING-GEAR CONNECTING RODS

Few automobile parts have been standardized to the same degree as the tie rods and drag links of steering gears. The Recommended Practice covering this subject relates entirely to the connectors at the ends

SOCKET



TYPE 1. DESIGN WITHOUT SPACER



TYPE 2. DESIGN WITH SPACER

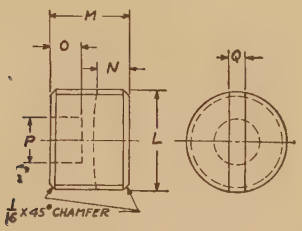
Nominal Ball Size	*A*	*B* Minimum	*C*	*D*	*E*
1	{1.005 / 1.015}	1-1/4	2-7/8	1-1/16 x 20	3/4
1-1/8	{1.127 / 1.132}	1-3/8	3-1/4	1-3/16 x 16	7/8
1-1/4	{1.250 / 1.255}	1-1/2	3-3/4	1-5/16 x 18	1-3/16
1-1/2	{1.505 / 1.515}	1-7/8	4-3/8	1-19/32 x 14	1-1/4
1-3/4	{1.785 / 1.795}	2-1/8	5	1-7/8 x 12	1-3/8

Nominal Ball Size	*F*	*G*	*H*	*J*	*K*
1	3/16	5/32	1.015	1/2	1-1/4
1-1/8	3/16	5/32	1.140	9/16	1-1/2
1-1/4	3/16	5/32	1.265	1	1-15/16
1-1/2	5/16	7/32	1.515	1	2
1-3/4	5/16	7/32	1.765	1-1/8	2-1/8

Tubing to be cold-drawn seamless steel not inferior to S. A. E. 1020.

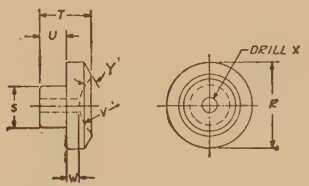
STEERING GEAR CONNECTING RODS—*Continued*

PLUG



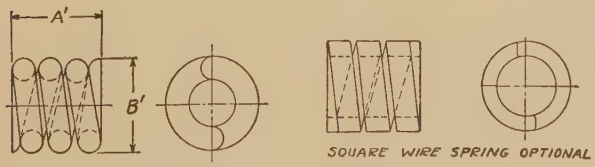
Nominal Ball Size	L	M + 1/64 —0	N Minimum	O + 1/64 —0	P Minimum	Q Minimum
1	1-1/16 x 20	21/32	23/64	13/64	7/16	5/32
1-1/8	1-3/16 x 16	3/4	3/8	17/64	13/32	5/32
1-1/4	1-5/16 x 18	1-1/16	13/32	13/32	5/8	5/32
1-1/2	1-19/32 x 14	1-1/8	1/2	15/32	57/64	1/4
1-3/4	1-7/8 x 12	1-1/4	1/2	17/32	1-1/32	1/4

BALL SEAT



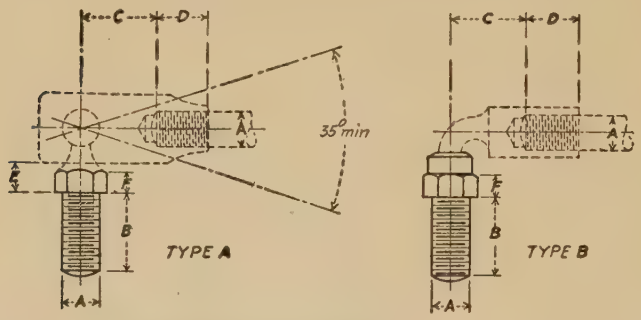
Nominal Ball Diameter	R —0.010 +0	S Maximum	T +0 —1/64	U +0 —1/64	V +0.005 —1/64	W +0 —0.005	X	Y
1	0.995	0.427	33/64	3/16	1/2	0.187	9/64	45°
1-1/8	1.115	0.557	37/64	1/4	9/16	0.187	3/16	45°
1-1/4	1.234	0.591	23/32	3/8	5/8	0.187	3/16	30°
1-1/2	1.478	0.843	15/16	7/16	3/4	0.312	1/4	45°
1-3/4	1.750	0.984	1-1/8	1/2	7/8	0.375	1/4	30°

SPRING



Nominal Ball Size	Approximate Free Length A'	Outside Diameter B'	Working Load, Lb.
1	7/8	31/32	275 = 25
1-1/8	1-1/16	1-3/32	375 = 50
1-1/4	1	1-3/16	450 = 50
1-1/2	1-5/32	1-15/32	400 = 50
1-3/4	1-1/4	1-11/16	580 = 50

S. A. E. Standard Ball-and-Socket Control Joints



A Diam.	A Threads Per Inch (NF-2)	Hex. Min.	B	C	D	E[1]	F[2] Min.
No. 5	40	1/4	9/32	1/4	1/4	7/32	3/32
No. 8	32	1/4	11/32	11/32	5/16	7/32	7/64
No. 10	32	5/16	13/32	13/32	3/8	1/4	1/8
No. 12	32	3/8	15/32	7/16	3/8	1/4	9/64
1/4	28	3/8	9/16	1/2	7/16	9/32	5/32
5/16	24	7/16	23/32	5/8	9/16	9/32	3/16
3/8	24	9/16	7/8	3/4	11/16	5/16	7/32
7/16	20	5/8	1	7/8	13/16	3/8	1/4
1/2	20	3/4	1 1/8	1	15/16	7/16	9/32

[1]Minimum wrench clearance.
[2]F = ½ A + 1/32.
Dimensions in inches.

S. A. E. Standard Brake-Lining Sizes

Width, In. ±1/16	Thickness, In.			
1 1/8	1 1/8	1 1/8		
1 1/4	1 1/4	1 1/4		
1 1/2	1 1/2	1 1/2		
1 5/8	1 5/8	1 5/8		
1 3/4	1 3/4	1 3/4	1 3/4	
1 7/8	1 7/8			
2	2	2	2	
2 1/4	...	2 1/4	2 1/4	
2 1/2	2 1/2	2 1/2	2 1/2	
2 3/4	...	2 3/4	2 3/4	
3	...	3	3	
3 1/4	...	3 1/4	3 1/4	
3 1/2	...	3 1/2	3 1/2	
3 3/4	...	...	3 3/4	
4	...	...	4	4

INDEX

Acceleration, 7
Acceleration curves for buses, 563
Accelerator pedal, 418
Accelerator switch, 560
Ackermann steering gear, 375
Air brakes, 346
Axle bearing thrust loads, 171
  housing thrust loads, 183
  housing, types of, 179
  housings, allowable stresses in, 183
  housings, pressed steel, 179
  shafts, 173
  torque calculation, 176
  tubes, calculation of, 260
  tubes, swinging, 503
Axles, cranked, 506

Ball and socket control joints, S. A. E., 672
Ball studs, 666
Balloon tire inflation table, 655
Balloon tire load table, 655
Band brake, Overland, 295
Band brakes, advantages of, 303
  details of, 295
  expanding, 301
  wrapping, 301
Bearing sizes, determination of, 208
Bevel-gear efficiency, 218
  bearing loads, 158
  blank calculation, 159
Bevel gears, center of load distribution in, 157
  strength of, 154
Belflex fabric spring shackles, 492
Belflex torque arm support, 216
Booster brakes, 342
Borg & Beck clutch, 14
Bowden wire mechanism, 415
Brake adjustment, 322
  at anchor, 324
  by cable, screw and nut, 324
  by two-part lever, 323
  by worm and wheel, 323
  Cadillac, 324
Brake adjuster, self-locking, 322
Brake adjuster, turnbuckle type, 322
Braking area required, 294
Brake band, distribution of tension in, 299
  cables, 321
  cams, floating, 307
  clearance, 326
  cross shaft, requirements in, 316
  control mechanism, Perrot, 332
  drums, 293
  drums, method of fastening, 513
  equalizers, 315
  expanding mechanism, toggle, 302
  hook-ups, 319
  lay-out for Hotchkiss drives, 329
Brake band, leverage factor, 326
  lever, emergency, 426
  lever mechanism, Bendix, 334
  lining materials, 290
  lining sizes, S. A. E. standard, 672
  lining tests, 292
  linings, friction, coefficients of, 291
  Lockheed hydraulic, 338
  operating mechanism, front-wheel, 332
  proportioners, 317
  retracting spring, 321
  rod adjuster, Ford, 322
  shoe linings, angular limits of, 309
  shoe pressure diagram, 309
  shoes, distribution of pressure in, 308
  shoes, methods of anchoring, 305
  types of, 281
Braking power, calculation of, 283
Brakes, 10, 280
  four-wheel, 330
  historical development of, 282
  legal requirements with respect to, 280
  metal-to-metal, 347
  number of, 283
  self-energizing, 306
  wrapping effect in, 296
Brinell test, 641
Browning, 637
Brown-Lipe clutch, 24
Budd disc wheel, 521
Buffalo corrugated taper drive, 519
Bumper pads, 447
Bumpers, commercial vehicle, 448
Bus design regulations, British, 656

Cadmium plating, 634
Cast steel for wheels, composition of, 527
Cast steel wheels, weight of, 527
Castering of front wheels, 364
Caterpillar tractor steering gear, 612
Causan shimmy damper, 408
Chain adjusting rods, 570
  and sprocket calculation, 567
  drive, 564
  drive, effect of spring-play on, 572
  links, offset, 567
  pull, 569
Chains, capacity of, 568
  construction of, 564
  for use with motor trucks, 566
Chamfering allowances, 80
Change-speed gear, 6, 67
  Levassor's, 68
  Maybach's, 70
  Renault's, 69
Change-speed gears, obsolete types of, 67

Chassis design, 2
Chassis lubrication by grease cups, 534
  metering units for, 540, 542
Chassis lubrication system, Alcyl, 546
  Alemite, 537
  Alemite automatic, 543
  Bijur, 541
  Bowen, 539
  centralized, 539
  high pressure, 537
  Myers', 535
  Tecalemit, 546
  Zerk, 538
Chassis lubricator fittings, 538
Chassis lubricators, Madison-Kipp, 536
Chassis springs, 453
Chromium plating, 633
Circular thickness, determination of, 64
Cletrac tractor steering gear, 613
Clutch adjustment, 24, 41
  brake, 31
  capacity, limit of, 19
  center, flexible, 33
  driven member, inertia of, 21
  direct-cone type, 35
  driving means, 39
  facings, S. A. E., 661
  gear-type of positive, 101
  Hele Shaw, 37
  insert, 33
  inverted conetype, 35
  jaw-type of positive, 101
  lever-operated, 39
  levers, compound, 21
  levers, flexible, 23
  lubrication, 28
  metal-disc, 37
  multiple dry disc, 38
  over-running, 119
  Packard, 39
  shaft, 30
  single-plate, 14
  springs, 29
  throw-out bearing, 26
  throw-out collar, 26
  throw-out sleeve, 27
  throw-out yoke, 20
  tractor type, 39
  Twin Disc, 39
Clutching, double, 32
Commutating poles, 554
Companion flange, 141
Cone clutches, calculation of, 36
Corrosion, principles of, 629
Cottin-Desgouttes suspension, 501
Crawler tractors, 602
  arrangement of, 604
  proportions of, 606
  requirements in, 603
  weight distribution in, 605
Crawler tractor, track carrier roller, 611
  track links, 608
  track pressure, 606
  track rollers, 610
  sprocket calculation, 607
  steering gear, 612
Critical speeds, 125
  analysis of, 127
Control, 410
  assembly, Gemmer, 412
  assembly, Reo, 413
  friction-lock, 411
  widths, 433
  joints, 416
  lay-out for passenger car, 429
  levers, location of, 423
Control, levers on steering post, 415
  parts at bottom of column, 414
  ratchet lock, 411
Cross springs, suspension by, 501

Daimler arched rear axle, 503
Dayton steel wheels, 527
Demountable wheel locking means, 518
Disc brakes, 312
Disc clutches, calculation of, 17
Disc wheels, 520
Differential carrier, 171, 189
Differential gear, 9, 151
  action of, 152
  details of, 162
  non-slip type, 165
  proportions of, 156
  single-piece-housing type, 162
  spur-type, 164
  two-pinion type, 162
Differential gears, number needed with four-wheel drive, 273
Differential lock, 166
Direct drive, 7
Double-reduction axle, 245
  Eaton, 245
  Mack, 248
  Renault, 250
  White, 249
Double-reduction axles, calculation of, 259
Double-roller bearings at axle ends, 259
Drag link lay-outs, 397
Driving wheels, 3
Dual disc wheels, 525
Duerkopp independent suspension, 503
Durant clutch, 22

Endurance limit, 644
  electric short-time test for, 644
  thermal short-time test for, 647
Electric braking, 560
Engine arm brackets, 440
Engine location, 5
Entz system, 548
Equalizers, disadvantages of, 316

Factor of safety, basis for, 125
Fatigue of metals, 644
Fender brackets, 449
Ferroxyl test, 639
Final drive ratios, 8
Final drive, types of, 168
Firestone demountable rim, 531
Fischer cast steel wheels, 528
Fits and tolerances, 615
Fits, basic-hole system of, 616
Fits, basic-shaft system of, 616
Fits, types of, 615
Frame, 5, 430
  channel section data, 436
  cross members, 438
  cross members, location of, 439
  materials, 430
  members, joints between, 443
  outriggers, 447
  parts, proportions of, 437
  sections, 431
  side rails, bending moments on, 436
  side rails, shearing forces in, 435
  side-rail sections, calculation of, 434
  types, 431
Frames, stresses on, 438
Four-shoe brake, Timken, 314
Four-wheel drive, 4, 265

Four-wheel drive drop gear, 267
Four-wheel drives, loss of power due to slippage in, 272
Free-wheel drives, 119
Friction, clutch, 6, 14
Friction coefficients of clutch linings, 17
Friction facings, 16
Front axle, 348
  bending moments on, 348
  center sections, 352
  I-section, 353
  spring pads, 355
  steering heads, 356
  torsional moment on, 350
Front wheel bearings, 366
Front wheel brake, automatic release of outside, 333
  Isotta-type, 335
  Pierce Arrow, 335
  requirements of, 331
  torque, 350
Front-wheel drive, 263
  arrangement of, 264
  British War Office, 268
  F. W. D., 268

Galvanized iron, 630
Gas-electric drive, 547
  circuit connections, 554
  efficiency of, 559
  Entz system, 548
  generator connections, 551
  generator ventilation, 558
  internal losses in, 557
  proportion of generator to engine capacity, 562
  relation between current and torque, 550
  relation between voltage and speed in, 550
  separate excitation, 553
  single vs. dual motor, 555
  Tilling-Stevens, 549
  weights of, 556
Gauges and gauging, 621
Gauging, principles of, 623
Gear calculations on basis of compression stresses, 82
  calculation, sample, 56
  carrier, 171, 189
  elements, proportions of, 45
  materials, compression strengths of, 51
  problems, graphical solution of, 651
  problems with internal gear transmissions, 115
  ratios, 75
  ratios, geometrical series of, 75
  shifting, operation of, 31
  shift lever, 424
  shift positions, 426
  speeds, numbers of, 7
  standards, metric, 66
Gear teeth, compressive stresses on, 49
  in contact, number of, 52
  minimum number without interference, 60
  pitting of, 47
  stub, 46
Gear-tooth diameter increments, 62
  interference, 59
  proportions, 56
  rubbing speed, 56
  strength, Lewis formula for, 47
Gearing, principles of, 42
Gears, cyanide-hardened, 73
  helical, 64, 651
  hypoid, 205
    long-and-short addendum, 58
  nitrided, 73
  strength of, 47
Generator drive, General Electric Co., 562

Hardness tests, 641
Hayes demountable rims, 531
Heat-treatment terminology, 643
Helical gears, rules for direction of helix, 66
Helper springs, 495
Hertz compression stress equation, 49
Hotchkiss drive, 212
Hupmobile shifter gear, 107
Hypoid gears, 205
  bearing loads with, 207

Indirect fourth speed, 210
Internal gear drive, details of, 257
International Harvester Company, Internal-gear drive axles, 255
  London General Omnibus Company, 255
Internal gears, 252
  location of, 253
  relative efficiency of, 252
Involute, the, 42

Jeantaud's diagram, 375
Joints, over-running, 321

Kelsey split rim, 533
Keyed shafts, failure of, 645
Knuckle-arm angles, proper, 382
Knuckle arms, 368
Knuckle pin bearings, loads on, 359
Knuckle spindle dimensions, 365
Krupp process of nitriding, 73

Lanchester shimmy damper, 408
Lancia front wheel suspension, 499
Larrard on torsional strength of splined shafts, 123
Leaf springs, theory of, 458
Lewis formula for gear tooth strength, 47
Lewis tooth-form factor, 49
Live axle, parts of, 169
Live axles, classification of, 169
Locking of wheels, objections to, 287
Long clutch, 22, 33
Lubrication, chassis, 534
Lubrication, Myers' magazine system, 535
Lubrication, Wick Feed, 535

Magnesium alloy wheels, 529
Malleable iron for wheels, composition of, 528
Materials, spring, 454
Materials, torsional strength of, 122
Materials for steering knuckles, 358
Metal bath coatings, 630
Michelin disc wheel, 520
Module, 66
Motor buses, wheelbases of, 12
Motorcycle, 1

N. A. G. independent suspension, 507
Nickel plate coating, thickness of, 632
Nickel plating, 630
Nickel plating, double-salt solution for, 631
Nickel plating, Packard process of, 631

Oil guards, 106
Owen Magnetic transmission, 548

Paints and enamels, 629
Parkerizing, 636
Passenger cars, wheelbases of, 12
Pedal pads, 423
Pedal pressure, permissible, clutch, 20
Pedals, adjustable, 421
Pedals, cluch, 20
Pedals, brake, 419
Pipe thread standard, 662
Pitch of gears, 43
Plow draft, 575
Power take-off, truck, 664
Pressure angle, 44
Propeller shaft intermediate bearing, 149, 663
Propeller shafts, 9, 121

Quickest stop, conditions ensuring, 285

Radius rods, 216
Radius rods, calculation of, 571
Railcar, 2
Rear axle, arched, 503
  brace rods, diagonal, 214
  center-drop type, 250
  center housing, roller-bearing, 189
  center housing, ball-bearing, 191
  drive gear assembly, 189
  end, full-floating type, 258
  ends, heavy-duty, 257
  lubrication, 190
  outer end, semi-floating, 192
  outer end, three-quarter-floating, 194
  thrust, 210
  torsion, 210
  truss, 209
  types of, 168
Rear axles, bearing loads of, 171
  full-floating, 169
  semi-floating, 169
  three-quarter-floating, 171
Reverse-gear arrangements, 103
Reverse-gear lock-out latch, 109
Rims, Collapsible, 531
  Demountable, 510, 531
  Drop center, 530
  Split, 532
Rim with self-locking side ring, 532
Road wheels, 509
Road wheels, number of, 2
Rockwell hardness test, 642
Roller chains, heavy series, 665
Rolls-Royce wire wheel hub, 518
Rubber auxiliary springs, 493
Rubber shock insulators, 491
Rubber spring bumpers, 493
Rudge-Whitworth wire wheel, 518
Running-board brackets, 447
Rust-proofing, methods of, 629

Safety factors for axle shafts, 176
Salt spray test, 638
Saturants for brake linings, 291
Schoop metal spray process, 637
Scleroscope hardness test, 642
Screw thread series, national coarse, 659
Screw thread series, national fine, 659
Section moduli of common sections, 658
Self-energizing effect, 306
Semi-floating axles, shafts of, 176
Servo brakes, 342
Servo brake, hydraulic type, 345
Serrated fittings, 668
Shaft loads with differential lock, 167
Shaft loads with non-slipping differential gear, 167
Shaft materials, 121
Sherardizing, 635
Shift-lever mounting, 110, 424
Shimmy dampers, 408
Shoe brake, Columbia internal, 324
Shoe brake, railroad-type of, 312
Shoe brakes, details of internal, 304
Shoe brakes, external, 311
Shore hardness test, 642
Six-wheeler, British War Office, 277
  Buessing, 277
  Duekopp, 505
  N. A. G., 507
Six-wheeler, Safeway, 274
  Scammell, 278
  with single driving axle, 278
Six-wheelers, 270
  advantages of, 270
  effect on road stresses of, 270
Skidding, cause of, 331
Sliding gear, history of the, 68
  progressive type, 68, 70
  selective type, 70
  shafts, squared and splined, 100
Sliding joints, 133
Smooth engagement, means to ensure, 22
Speedometer drive, 105
Spicer on torsional strength of splined shafts, 123
Spiral bevel gears, 194
  bearing loads with, 200
  calculation of blanks, 196
  capacity of, 199
  direction of thrust loads on, 199
  minimum number of teeth with, 195
Splined joints, S. A. E. standard, 660
Spoke assembly for wood wheels, 513
Spoon-latch lever, 427
Spring bolts, 487
  brackets, 444
  camber, 467
  center bolt, 479
  center hump, Tuthill, 480
  deflection, 460
  deflection, effect of clamped portion on, 459
  distortion due to torque, 472
  eyes, 487
  flexibility, 470
  formula, application of, 473
  leaf alignment, 485
  materials, 454
  nip, 468
  periodicity, 464
  plate finish, 497
  plate gauges, 457
  plate stress tables, 462
  plates, concave, 497
  plates, number of, 463
  points, 486
  pressure-block, 483
  rebound clips, 484
  reverse leaves, 484
  seats and clips, 481
  shackle, Tryon self-adjusting, 489
  shackles, 487
  standard widths of, 457
  steel, endurance limit of, 456, 648
  stress calculation, 461
  stress due to braking, 471
  vibrations, frequency of, 464
  without center bolt, Woodhead, 480

Springs, average lengths of, 469
cantilever, 478
classification of, 453
eccentrated, 477
factors of safety for, 470
interleaf friction in, 496
for motor trucks, dimensions of, 471
with flat bearing surfaces, 489
with variable rate, 494
with wear plates, 490
Sprocket wheels, design of, 567
Spur-bevel-gear drive, 248
Steeldraulic brake, Midland, 337
Steel, medium carbon, 123
Steels, carburizing, 649
case-hardening, 71
normal and abnormal, 649
oil-hardening, 72
for axle shafts, 173
Steering, right vs. left, 384
arm proportions, 398
connectors, 396
drag link, 396
Steering gear, 375
cam-and-lever, 392
general arrangement of, 383
mounting of, 395
screw and nut, 389
split-nut-type, 391
types of, 385
worm and sector, 387
worm and wheel, 387
connecting rods, 670
parts, dimensioning, 386
adjustable for slope, 395
requirements, 385
safety in, 401
Steering head, Reverse Elliott, 356
mechanism, theory of, 376
of six-wheelers, 404
knuckle in hub form, 358
parts, strength of, 369
pivots, inclined, 361
problem, analytical solution of, 379
problem, graphical solution of, 376
shaft, vertical, 394
stops, 374
wheel camber, 362
wheel gather, 363
wheel rubber rims, 401
wheel toe-in, 363
wheels, 3, 399
Stopping distances, 284
Straddle-mounting of driving pinion, 191
Stub teeth, 46
Suspension, independent, 498
Suspension, pneumatic, 506

Tatra independent suspension, 503
Taxicab, 1
Tenon sizes for wood spokes, 511
Terne plate, 630
Three-shoe brake, Bendix-Perrot, 341
Thumb-latch lever, 427
Tie rod, 371
Tie-rod connectors, 372
Tire carriers, 452
Tire pump mounting, truck, 664
Tire rims, 531
slip, 547
Tires, 10
clincher, 530
straight side, 530
types of pneumatic, 530
Tolerances, American standard on, 618
bilateral, 616
Tolerances, for automobile parts, 623
unilateral, 616
Torque arms, 214
Torque multiplication, 7
Torque tubes, 212
Torsional strength of chrome-nickel steel, 124
of chrome-molybdenum steel, 124
of medium carbon steel, 123
Tooth load, distribution of, 58
Tooth modification, 55
Tooth stresses, allowable, 79
Tractor belt pulleys, 594
belt speeds, 594
clutches, 580
design, 575
drawbars, 597
front axles, 593
gear reduction ratios, 581
gears, dimensions of, 589
hitch, point of, 596
horse power (drawbar), 576
drive, arrangement of, 582
industrial, 1
power take-off, 600
Tractor rear axles, 591
general arrangement of, 579
rearing tendency of, 577
side draft on, 594
speeds, 576
steering gears, 593
tests, Nebraska, 575
transmission, three-speed, 584
transmission, Fordson, 587
transmission, McCormick-Deering, 584
truck, 1
wheelbases of, 13
wheel lugs, 600
wheels, 599
wheel sizes, 598
weight and drawbar pull, 577
Transfer of weight due to driving torque, 264
due to braking, 287
Transmission bearing loads, 86
bearing mountings, 99
bearing sizes, 96
Cotta, 117
direct-drive clutch, 100
drive-shaft mounting, 99
housings, 103
gear calculation, sample, 81
gears, choice of pitches for, 78
gears, layout of, 76
gears, materials for, 70
gears, tooth numbers for, 77
G. M. T., 115
lock-out latch, 109
lubrication, 106
reverse gear arrangements, 103
secondary shaft assembly, 102
sliding mechanism, 107
shaft dimensions, 98
Warner Hi-Flex, 113
with internal gears, 111
Transmissions, constant-mesh, 117
efficiency of, 119
gear ratios of, 75
heavy-duty, 117
multiple-speed, 115
Tread, 11
Trucks, wheelbases of, 13
Truck transmission calculation, 83
Turning radius, 402
of trailer, 403

Udylite process of cadmium plating, 635
Undercut, pinion diameter enlargement to avoid, 62
Unequal-sized wheels, effects of, 271
Unit pressure, allowable, 19
Universal joint, 9, 134
  Blood Brothers, 144
  Cleveland, 145
  constant velocity type, 146
  Mechanics Machine Co., 144
  pot-type of, 142
  Spicer, 143
  Spicer-Hardy, 148
  Weiss, 146
Universal joints, capacity of disc-type, 149
  details of design of, 140
  disc-type, 147
  lubrication of, 143
  mathematical theory of, 136
  metal-centered disc, type, 148
  methods of compensating for inequality of transmission of, 140
  oil-lubricated, 144
  pot- and cross-types compared, 142
  speed fluctuations in, 135

Vacuum brake, 343
Vehicles, types of, 1
Ventilation, generator, 558
Vitreous enamel coatings, 635
Vomag bus suspension, 507

Weight transfer, 3, 264, 287
Wemp, E. E., on clutches, 23, 26
Westinghouse vacuum brake, 343
Wheelbase, 11
Wheel bearings, 209
Wheel brakes, connections to, 327
Wheel diameters, 509
Wheel shimmy, 405
Wheel wobble, 405
Wheels cast, 525
  demountable, 510
  interchangeable, 523
  magnesium, 529
  malleable iron, 528
  tests of by Bureau of Standards, 525
  truck-type wood, 512
  wood, 511
Wire spokes, 517
Wire wheel, bolted-on type, 519
Wire Wheel Corp. of America wire wheels, 519
Wire wheel driving means, 518
Wire wheels, 515
  Ford, 520
  methods of lacing, 516
Wood wheel, demountable, 522
Wood wheels, advantages of, 513
Worm drive, 219
Worm drive, advantages of, 219
Worm-driven bus axle, 239
Worm-gear bearing loads, 229
  bronzes, 220
  center distance, 225
  drives, capacity of, 225
  efficiency, 228, 241
  pressure angles, 223
  testing machine, Lanchester, 241
Worm gearing, theory of, 222
Worm-gears, FJ-type, 233
  Hindley, 232
  hourglass-type, 232
  mounting of, 235
  proportions of, 226

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